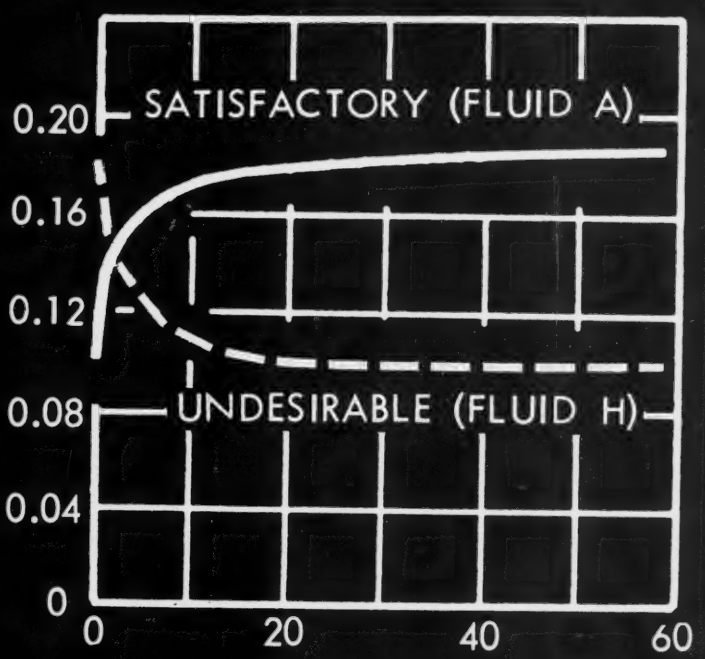
Vol. 17, No. 3 March, 1961

LUBRICATION ENGINEERING

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE



See page 110 for...

Automatic Transmission Fluids

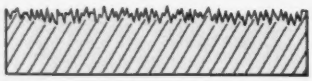
JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

CLIMAX MOLYBDENUM

Molysulfide® News Digest

CLIMAX MOLYBDENUM COMPANY, a division of American Metal Climax, Inc., 1270 Avenue of the Americas, New York 20, N.Y.

MOLYSULFIDE INCREASES TOOL LIFE AS MUCH AS 300 PER CENT



When magnified, a piece of highly polished metal is actually smooth to touch, yet presents a cross-section of saw-tooth irregularity. Molysulfide fills in the valleys, prevents the peaks from tearing, permits plastic deformation of the metal to make a smoother surface.

COMPOUND CONTAINING MOLYSULFIDE REDUCES MACHINERY WEAR-IN TIME

During the wear-in period of machinery, permanent surface damage . . . variously described as *galling, scuffing, scoring, tearing, scratching, excessive abrasion* and *seizing* . . . is an inherent hazard.

After years of experience, leading equipment builders agree that an MoS_2 compound is the best guarantee against wear-in damage. Here are five ways such a compound, applied to moving parts prior to operation, can save you wear-in time, money and possible grief:

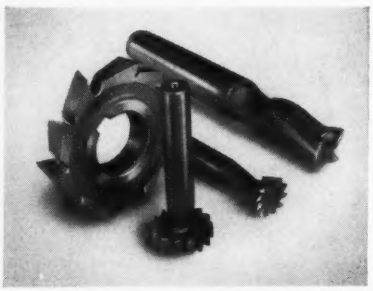
1. Shortens wear-in period without use of abrasives.
2. Eliminates stick-slip behavior and resists galling and seizing at bearing pressures far beyond the yield point of any metals.
3. MoS_2 particles have a tenacious adherence to bearing surfaces. Will not scrape off.
4. Need not be burnished into surface. Brush or wipe on a thin coating.
5. Inexpensive in the long run. A little goes a long way.

When writing, refer to CL-103

MoS_2 Provides Greater Resistance to Wear at High Temperatures

The Alpha-Molykote Corporation in Stamford, Conn., is now marketing a new Molysulfide bonded lubricant for ferrous surfaces.

Applied with a conventional paint spray gun, this compound fills the microscopic valleys of the surfaces of tool steel and gives on-the-spot lubrication that allows the chips to glide. Once sprayed, it dries at room temperature to a tough film that has a thickness of 0.0001—0.0008 in. and a coefficient of friction of 0.03 at 100,000 psi.



Extremely resistant to wear at high pressures and temperatures (operating temperature range -300 to 600 F) this new compound requires neither chemical nor mechanical surface pretreatment and, because of its high adherence and low friction coefficient, it increases cutting tool life as much as 300%.

Effectiveness of this lubricant comes mainly from the weak layer-to-layer bonds which allow easy sliding and, within individual layers, from the cohesive forces between Mo and S atoms that are quite strong and prevent penetration by surface asperities.

Minimum coverage coating is 285 sq. ft./lb., and consumption is about 1% of tool weight. Normal lubricating oils, cutting fluids or solvents have no effect on these coatings.

When writing, refer to CL-101

Punch Life Increased from 2,000 to 25,000 Holes in Steel Plate

These punches (11/16") were used to punch 25,000 holes in ½"—1020 cold rolled steel plate. Without a lubricant containing MoS_2, a maximum life of only 2,000 holes could be obtained.

A West Coast manufacturer was experiencing low punch life with the best available lubricants. His multiple punching operation consisted of cold shearing 50 holes simultaneously in a single steel plate. A maximum life of 2,000 holes per punch was obtained with ordinary lubricants.

Then, a compound containing Molysulfide was applied to the punch. The result: punch life was increased from 2,000 to 25,000 holes and, as can well be imagined, considerable savings were effected.

When writing, refer to CL-102

Volume 17, No. 3
March, 1961

Lubrication Engineering

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

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All general correspondence, editorial correspondence, and correspondence concerning advertising and subscriptions should be addressed to The American Society of Lubrication Engineers, 5 North Wabash Ave., Chicago 2, Illinois.

LUBRICATION ENGINEERING is published monthly by the American Society of Lubrication Engineers, 404 N. Wesley Ave., Mount Morris, Illinois.

Current volume single copies $1.00, copies to foreign addresses add 50¢ per copy. Photoprints of articles prior to the current volume may be obtained from the John Crerar Library, 86 East Randolph St., Chicago 1, Ill. Annual subscription of $6.00 in the USA and Canada, $7.00 to all other countries. Copyright by the American Society of Lubrication Engineers, 1961. ASLE will not be responsible for statements or opinions presented at its meetings or printed in its publications. Second class postage paid at Mount Morris, Ill.

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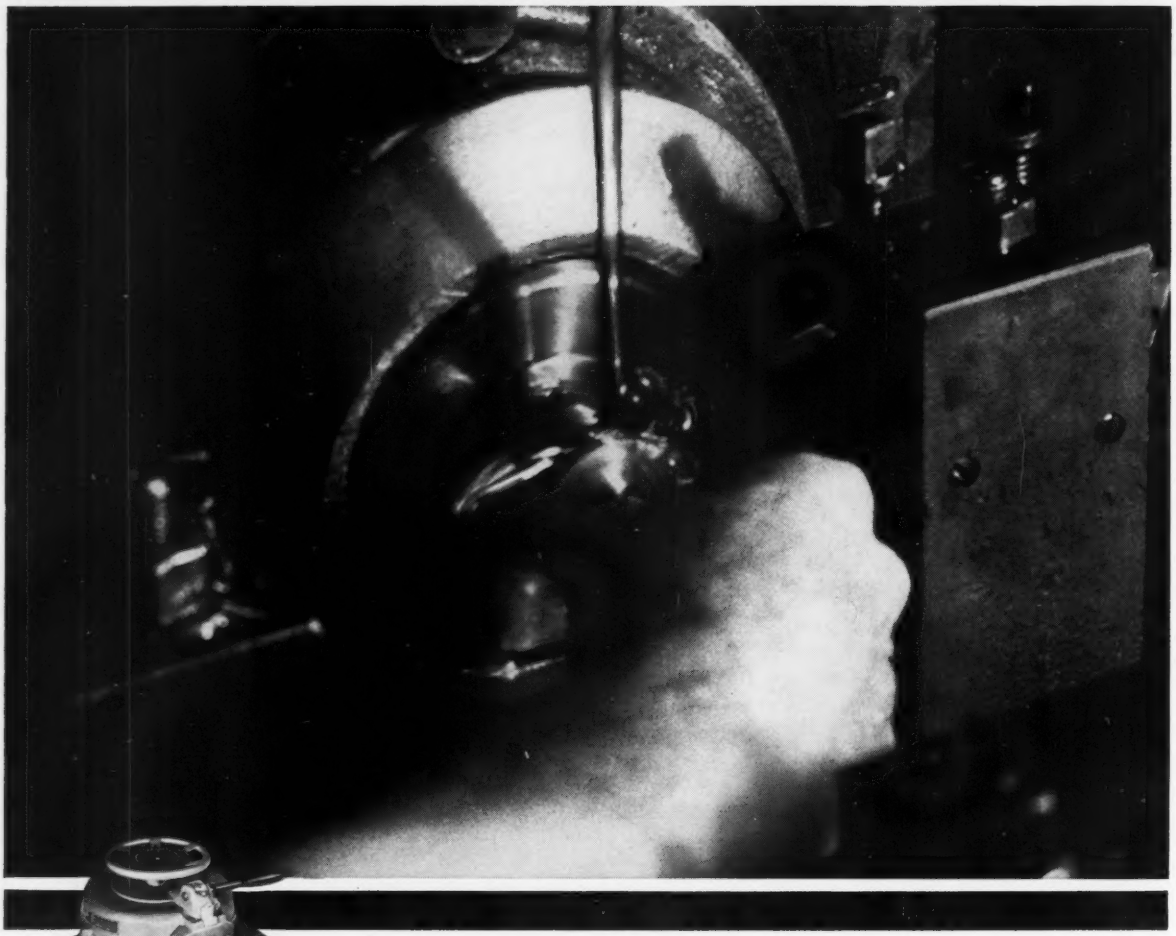

COOL CUTS

Keeping close tolerances is of prime importance at Diagraph-Bradley Industries, Inc., of Herrin, Illinois, leading manufacturer of stencil machines and accessories. One of the ways this is accomplished is by keeping cutting edges cool.

At Diagraph-Bradley stencil punches must be cut and held within .0005 of an inch—a fine point of precision Diagraph-Bradley attributes to the cooling and lubricating effect of Cities Service Chillo 10 cutting oil.

"You have only to take a Diagraph-Bradley machine," says one of the company officials, "cut a stencil with it and look at the resulting letter to realize how important the right cutting oil is to our production."

Cities Service Chillo 10 oil is a sulphochlorinated oil. The concentrations of sulphur and chlorine are stabilized by a special process, assuring uniformity of action through the entire cutting operation.

That's why, with Cities Service Chillo 10 cutting oil, work is always more accurate, clean and cool!

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QUALITY PETROLEUM PRODUCTS

LUBRICATION IN THE NEWS

The Society will not be responsible for the accuracy of statements made in this column. Any reference to proprietary products does not imply an endorsement of such products by the Society.

ARMY ENGINEERS DEVELOPING STANDARD FILTER/SEPARATORS

A family of standard filter/separators in 50, 300 and 600 gpm sizes for decontaminating the hydrocarbon fuels used in military aircraft and vehicles is being developed by the U. S. Army Engineer Research and Development Laboratories, Fort Belvoir, Virginia.

Although commercially available filter/separators may be modified and improved to meet necessary military requirements, this equipment varies considerably in configuration and dimensional details. This variation prevents the interchange of solids and water removal media of any one manufacturer within the vessel of another manufacturer, thereby creating an undesirable parts replacement problem in the field which could become critical during wartime.

In the standard design, dimensions of the filter/separator vessel, the solids and water removal media, and the media mounting will be specified, thereby insuring that the media of every approved manufacturer always fit into vessels manufactured by other firms in conformance with government specifications.

Experimental models of standard filter/separators in the three sizes are currently being evaluated in the Laboratories' test facility. Preliminary test results have been satisfactory, and 17 additional units have been ordered for delivery this year to appropriate military units for evaluation under field conditions. Type classification of the standard design is anticipated during Fiscal Year 1963.

The standard filter/separators were designed by the Laboratories in conjunction with Armour Research Foundation, Chicago, Illinois. (Source: U. S. Army Engineer Research and Development Laboratories, Fort Belvoir, Va.)

EXTENDED LIFESPAN OF SILICONE OILS REPORTED

The useful life of commercial silicone oils can be extended from three to five times through the use of oxidation inhibitors effective in other systems, according to a Navy study of the stabilization of silicone oils at temperatures between 250 and 371 C. A research report of this study has been released to science and industry through the Office of Technical Services, Business and Defense Services Administration, U.S. Department of Commerce, Washington 25, D. C.

Also available from OTS are a study of the lubrication behavior of liquid metals and an Air Force report on the effect of varying degrees of humidity on the consistency of 22 greases in storage. The three reports are described below:

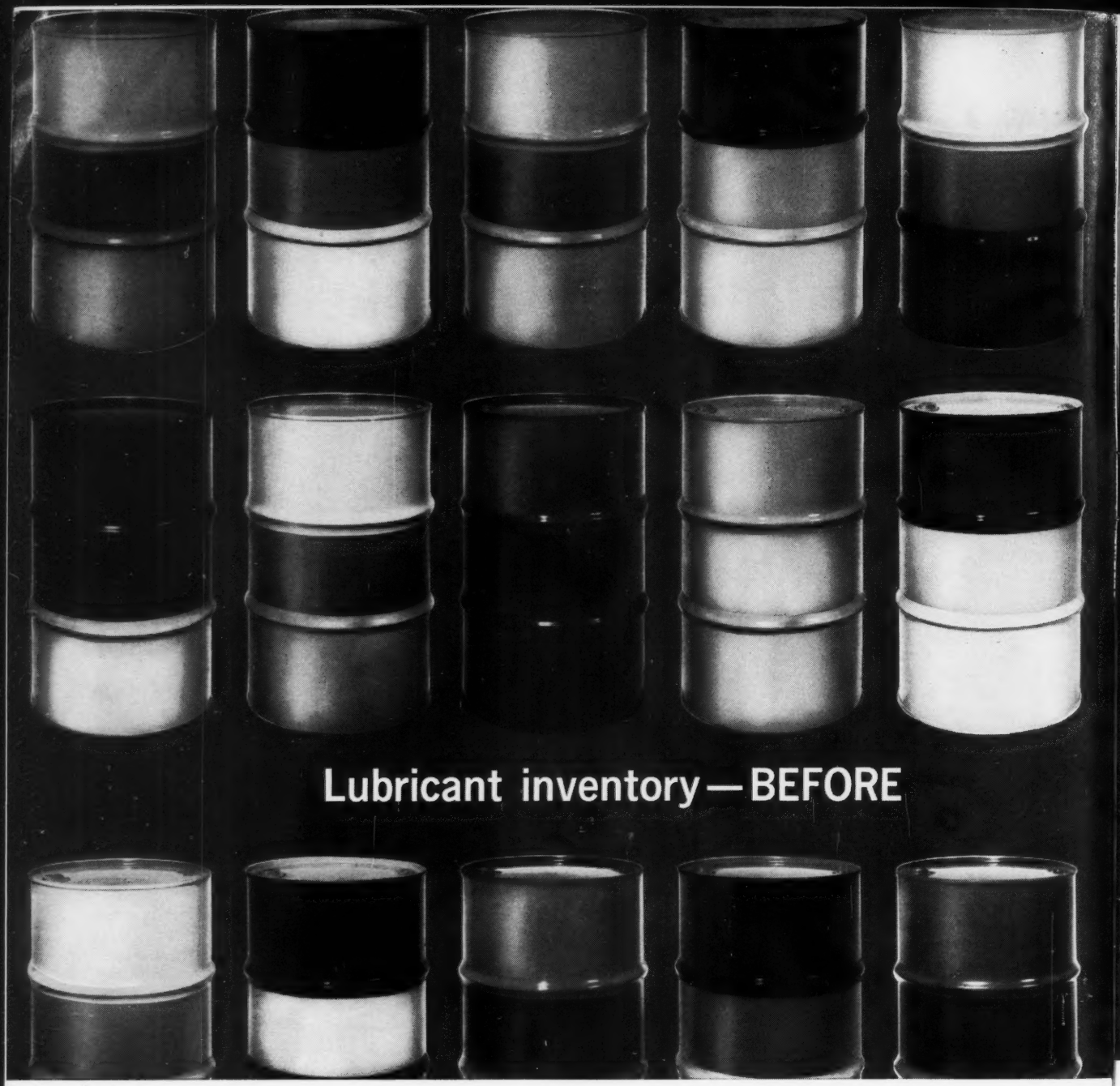
The Stabilization of Silicone Lubricating Liquids Above 200 C: Part 1—Conventional Oxidation Inhibitors and Some High-Surface Solids. H. R. Baker and others, U.S. Naval Research Laboratory. June 1960. 33 pages. (Order PB 161 314 from OTS, U.S. Department of Commerce, Washington 25, D. C., price $1.) Faced with the problem of ball bearing lubricating oil breaking down at temperatures over 200°C., the Naval Research Lab has found a way to extend the useful life of commercial silicone oils from three to five times by using oxidation inhibitors which have proved effective in other instances. This silicone oil stabilization study was made at temperatures between 250°C and 371°C where deterioration of silicones is primarily an oxidation process. The study showed that oxidation was inhibited by high-surface solids such as Shawanigan black, copper phthalocyanine, and ferric oxide, by conventional inhibitors of the arylamine type, and by dilauryl selenide. Dilauryl selenide was ineffective at 300°C and higher, whereas phenylaphanaphty lamine caused a relatively uniform improvement at 250°C and 300°C. The gains from any inhibitor were small at 371°C. The chemical diversity of the effective inhibitors—plus the relatively high temperatures at which they are effective—suggest that the mechanism inhibition in silicone oils differ from that of conventional oils at temperatures below 200°C. Research is continuing on the development of new types of inhibitors specifically for use with silicone liquids.

Lubrication Behavior of Liquid Metals. P. H. McDonald, North Carolina State College, for Wright Air Development Division, U.S. Air Force. March 1960. 76 pages. (Order PB 161 888 from OTS, U.S. Department of Commerce, Washington 25, D. C., price $2.) An experimental device has been designed and built to confirm the Reynolds equation theory for hydrodynamic lubrication in relation to liquid metals. This elaborate device was developed as part of an Air Force study of the lubricating behavior of liquid metals in both hydrodynamic and boundary lubrication. The report notes that because of the peculiar properties of liquid metals, a much closer correlation between theory and experiment can be anticipated than is usually obtained with conventional lubricants. A macrosconic theory of the boundary lubrication region was applied to a cylinder-flat combination. Tests showed that the lubricating behavior depended on the state of contact stress for the shape involved.

The Effect of Humidity on the Consistency of Greases. M. T. Fisher, Rock Island Arsenal, Department of the Army. April 1960. 10 pages. (Order PB 161 861 from OTS, U.S. Department of Commerce, Washington 25, D. C., price 50 cents.) To determine the effect of varying degrees of humidity on the consistency of 22 samples representing 13 different kinds of grease, the Army's Rock Island Arsenal conducted storage tests at relative humidities of 0, 50 and 100 percent for periods of up to seven months. The study resulted in the recommendation that before either soap- or nonsoap-thickened greases are stored or used under extreme high- or low-humidity conditions, their static and shear stabilities should be measured to determine if they are suitable for such service. The report points out that soap-thickened mineral oil greases are more likely to be stable in a moist atmosphere than soap-thickened synthetic fluid or nonsoap-thickened greases (Source: OTS, United States Department of Commerce.)

SAE ELECTS NEW PRESIDENT

At the termination of the Society's Annual Business Meeting on January 9, 1961, Dr. A. A. Kucher, Vice-President, Engineering and Research, Ford Motor Company, Engineering Staff, Dearborn, Michigan, took office as President of the Society of Automotive Engineers, Inc. for 1961, to succeed Mr. Harry E. Cheesebrough of Chrysler Corporation. (Source: Society of Automotive Engineers, Inc.)



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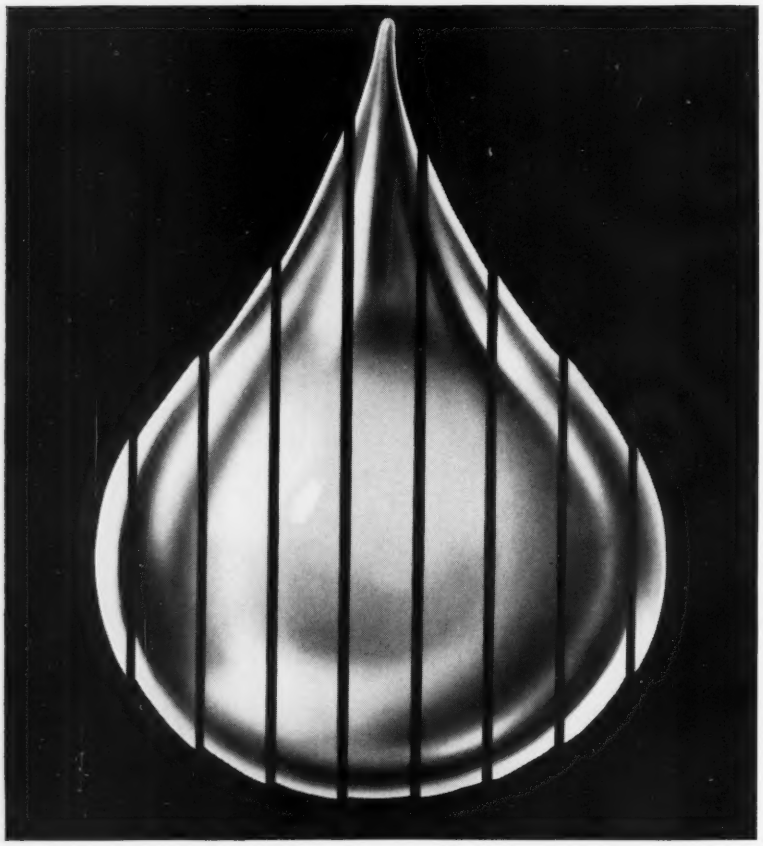

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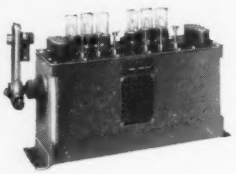
Whether you want to deliver several drops or the minutest fraction of a drop per piston stroke, a Manzel lubricator will do the job exactly. Manzel lubricators force oil of any viscosity against the high steam, gas and air pressure so common in modern compressors, engines and machines. They start, stop, speed up or slow down in synchronization with your equipment. Write for our catalog explaining the whole line. Manzel, 254 Babcock Street, Buffalo 10, New York. For efficient lubrication



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A UNIT OF HOUDAILLE INDUSTRIES, INC.

Manzel

SPECIALISTS IN LUBRICATORS AND METERING PUMPS SINCE 1898

INDUSTRY NOTES

SKF Industries, Inc., ball and roller bearing manufacturer, is proposing to erect a two and one-half million dollar research laboratory on about 25 acres in Radnor Township, Pa. Tentative plans call for breaking ground and completing the building within a year to 18 months.

Formation of Precision Metal Processing Company, incorporating as a Division, Kold-Lube Coaters Inc., has been announced by the Cleveland-based firm.

The expanded Precision Metal Processing Company offers complete metal processing facilities for manufacturers, die makers, fabricators and metal stampers.

Miniature Precision Bearing, Inc., Keene, N. H., manufacturer of miniature ball bearings, has completed an agreement with a French manufacturer of large size precision ball bearings, broadening the line which each company offers in its respective market area.

Miniature Precision Bearings, Inc., Keene, N. H., is now exclusive representative in the U. S. of ball bearings manufactured by Societe Nouvelle de Fabrications Aeronautiques (SNFA). At the same time, SNFA has sole representation in France for miniature ball bearings manufactured by MPB.

BULK GREASE DELIVERY

Grease-carrying tank trucks, which flow their cargoes into containers at industrial plants of customers, have made their bow in the Pittsburgh, Pa. district.

The Humble Oil and Refining Company, has begun the bulk delivery of lubricants to the Midland plant of Crucible Steel Company.

This plant is the only grease user in the area to accept truck flow delivery, as a result of having installed a flow system, including receiving tanks, at its plant.

After giving consideration to economies resulting from price, handling and elimination of waste, there is expected to be a minimum 40 per cent saving over conventional lubricant handling methods.

SECTION NEWS

Detroit—All Past Presidents of the Detroit Section were invited as guests to a Dinner Meeting preceded by a sponsored cocktail hour on January 24, 1961.

Mr. Willard D. Cheek, Sr. Research Physicist, GMC Research Laboratories, presented a very interesting discussion titled, "Fission, Fusion, and Confusion." After clearing much confusion and popular misunderstanding concerning fission and fusion, Mr. Cheek demonstrated numerous methods of employing these "tools" in laboratory investigations of wear, friction, and lubrication. He pointed out the value and application for research and industrial control laboratories.

The evening was climaxed by the presentation of a certificate of appreciation to each of the past presidents for his valued contribution and service to the society. (*See photo on page* 138)

Chicago—April 20, 1961 has been scheduled as election night. Mr. Arthur Stock, Acheson Colloids Co., will discuss "Solid Lubricants and New Ideas on Dry Films."

Buffalo—ASLE President, Mr. L. O. Witzenburg, spoke to a joint meeting of the Buffalo and Rochester Sections on March 15. His subject was "Worm Gear Lubrication."

Dayton—Mr. Robert Morgan, Garlock Packing Company, was the January speaker. His topic was "Bearings Capable of Dry Operation." Attendance was estimated at an all time high for a regular meeting. With "Making Membership More Desirable" as this year's aim, the Dayton Section has increased service to members by issuing abstracts of each talk, or pertinent literature on the subject of each meeting. In addition, speakers have been requested to stress money-saving features.

Montreal—Section members will tour Pratt and Whitney Aircraft Company Limited, Longueiul, Quebec, on April 5, 1961.

Cleveland – March 28, 1961 — Panel Meeting—subject: "Quenching Oils". Panel members will discuss practical aspects of quenching medium, testing and evaluation and quenching power. Jack Erickson, Chief Metallurgist, Euclid Div., GMC, will serve as moderator.

Pittsburgh—Mr. E. Crankshaw, Chief Engineer at Cleveland Graphite Bronze Co., presented the subject, "Bearing Materials and Applications" at the February meeting. Also at this meeting, Mr. M. M. Gurgo, ASLE Regional Vice-President, served as moderator for a panel discussion on industrial lubricants. The panel consisted of Mr. E. S. Reynolds, Staff Engineer, Socony Mobil Oil Co.; Mr. J. D. Lykins, Lubrication Engineer, Wheeling Steel Corp.; and Mr. L. B. Sargent, Aluminum Co. of America.

Oklahoma—Mr. Charles Bailey, Lubrication Engineer of U. S. Steel Corp., in his discussion of "Industrial Lubrication" at the February 24 meeting, covered the test methods used by U. S. Steel. He had on display their Mobil Training Unit.

Twin Cities—On March 3, 1961, the Twin Cities Section sponsored a one-way Education Program. The subject was "Hydraulics, Hydraulic System Components, and Hydraulic Fluids.

Hudson-Mohawk—The program of the April 20, 1961 meeting will feature Professor E. Rabinowicz, MIT. His subject will be, "Wear Processes."

Pictured above is Mr. L. O. Witzenburg, ASLE President and Mr. C. E. Genovese, U. S. Steel Corp. at the January 16 Meeting of the Chicago Section. Mr. Witzenburg talked on the progress and plans for ASLE and Mr. Genovese spoke on "How Management Looks and Lubrication."

NEW AND EXCLUSIVE with McCord Model 55 Lubricator

SAFE-LEVEL SIGHT FEED

That's right! The finest lubricator on the market today has been refined to give you even *more* for your money. Now, the McCord Model 55 Lubricator comes equipped with new, exclusive Safe-Level Sight Feed with these outstanding benefits for you . . . safe, sure warning of malfunction . . . quicker priming . . . better feed regulation . . . extreme accuracy . . . improved reliability. IMPORTANT: Safe-Level Sight Feed can also be added to existing Model 55 Lubricators.

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ELECTS LIFE MEMBER

The Board of Directors announces the election of Commander H. A. Shawk, USNR (Ret.) as a Life Member of the Society. With Commander Shawk's permission portions of correspondence relating to this matter are quoted.

Gentlemen:

I have been retired since 1953. I am now 80 years of age and do not expect to re-enter the field, as a practicing Professional Engineer, therefore I most respectfully request that this notification be considered by the Membership Board as a formal resignation, as a full member, from the ASLE.

I was Chief Lubrication Engineer, for 36 years, for the Associated Oil Co. of San Francisco, later the Tidewater Associated Oil, now Tidewater Oil Co. At one time I had 36 Lubrication Engineers on the staff, and, a wide circle of friends, covering the U.S.A. and abroad.

I always enjoyed my work and my membership in the ASLE and it is with deep regret that I submit this request. Please notify me when the Membership Board acts upon my request. Thanking you in advance for your consideration, I beg to remain,

Most sincerely yours,

/s/ H. A. SHAWK
Commander, USNR (Ret.)

Commander Shawk was advised of the recently adopted grade of Life Member in the Society and came back with this response.

Gentlemen:

I cannot pass up your kind offer of Life Membership established for us, "old re-treads". So long as I can write out checks (one-half rate dues) I will remain a member of the ASLE. I was always proud to be classed as a Lube Engineer, as I consider it a professio n which brings the engineer in close contact with his client, and opens up a broad field of sales, and efficient service to the industrial, marine, railroad, and automotive world.

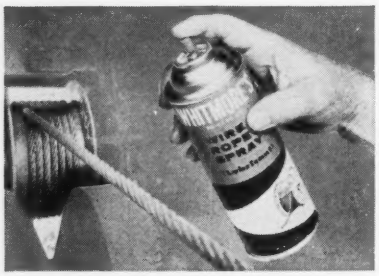
I hold unlimited Chief Engineer's license, "for Oceans", both steam and diesel, besides my naval service as an Engineer Officer, in three wars. For 54 years I was engaged in the practical operation of prime movers, and the application of Lubricants and fuels. This is a great profession.

Most respectfully yours,
/s/ H. A. SHAWK

Editors note: All hands should stand at attention and render a twenty-one gun salute to Commander Shawk.

TRIAL SAMPLES

Of World-Famous WHITMORE'S Lubricants NOW IN AEROSOL SPRAY-ON CANS!



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For open gears, sliding surfaces—exclusive formulas eliminate metal to metal contact, keep wear on the lubricant not the metal —no breakdown even after prolonged use under water—available for every climatic condition—packaged in handy 16-oz. aerosol spray-on containers or in bulk containers—send for a free trial sample.

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Exclusive formulas for lubricating and protecting wire rope, chain, springs. Penetrates to the core of wire rope minimizing internal friction and increasing usable life up to 300% —special protective qualities absolutely eliminate corrosion—non-gumming qualities reduce "carry-back"—packaged in handy 16 oz. aerosol spray-on containers or in bulk containers—send for a free trial sample.

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- Roller, Ball, and Sleeve Bearings
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ASLE ELECTS NEW BOARD MEMBER

Mr. John L. Finkelmann has been elected to the ASLE Board of Directors to fill the unexpired term of R. L. Leslie, Vickers Inc., who resigned. Mr. Finkelmann is associated with the Cycleweld Division, Chrysler Corp., Trenton, Michigan, and will serve as Central Regional Vice-President until April 1962. He has held many local and national offices in the Society Administration and his last position was as ASLE Secretary.

J. F. FINKELMANN

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Highlights of 1961 Annual Meeting Technical Sessions

by R. K. Gould, Program Chairman

The program for the 1961 ASLE Annual Meeting has been designed to "have something for everyone." Nineteen sessions with more than 70 papers and 3 panel discussions have been scheduled, and whatever your area of interest may be, as long as it relates to some phase of lubrication, there is bound to be something on the program for you—something you and your company absolutely cannot afford to miss. Here is a preview of the program.

Bearings and Bearing Lubrication

Five separate sessions will be held, including two on rolling contacts, one on gas lubricated bearings, one on bearings for heavy industry, and for the aviation people, one on airframe bearings. Never before has such a wide variety of papers on this important subject been presented at an ASLE Meeting.

Gears

The one session on gears will have papers on fretting wear, rocket engine gear boxes, and a method for determining gear oil stability.

Seals and Packings

Increased attention must be given to seals and packings because conditions to which these materials are being exposed are becoming more and more severe. There will be one session devoted to this subject which will include papers on metal bellows seals, mechanical carbon graphite, and seals for high temperature (600-1200°F.) applications.

Properties of Lubricants

Papers in this category are always of interest, and two sessions have been scheduled, one dealing specifically with low temperature properties of motor oils and the other being a general session with papers on high temperature anti-oxidants, mechanism of boundary lubrication with sulfur compounds, lubricants for instrument bearings, and a new grease equation.

Lubrication Fundamentals

This is another subject which is always popular, and there will be one large session which will include a group of papers on viscoelastic effects in lubricants, plus papers on abrasive wear resistance of bearing steels, lubrication of coated titanium, and others.

Engine Lubrication

For those interested in engine lubrication, this program has much to offer. In addition to the papers on low temperature properties of motor oils mentioned above, there will be two additional sessions, one dealing mainly with factors affecting engine wear and the other including papers on the lubrication of large gas engines and air cooled 2-cycle engines.

Fluids for Metal Working

The one session scheduled on this subject promises to be outstanding. In addition to papers on cutting fluid properties and quenching oils, there will be a symposium on lubricants for machining stainless steel including two papers and a panel discussion.

Lubrication Equipment

One of the most important in the series of steps leading to successful lubrication is proper application of the lubricant. This vital subject is covered in one giant sized session of seven papers, each on a different but related topic.

Steel Industry

There will be two sessions on lubrication in the steel industry. One will be a general session of three papers, including one on fire resistant hydraulic fluids. The other will be a panel discussion on steel mill lubrication in which an eight-membered panel will discuss such topics as testing of lubricants, bearing design, properties of greases and oils, gear lubrication, bulk grease handling, lubrication programs, automatic application and seals and packings. This should be of tremendous interest not only to steel mill representatives but also to Lubrication Engineers throughout all industry.

Machine Tool Lubrication

A session is scheduled on this subject and, among others, will include papers on organizing a plant lubrication program and the controversial matter of specifications of lubricants.

Railroad Industry

For those people connected with the railroad industry, a particularly interesting session has been arranged. There will be a paper on the use of greases in journal roller bearings and a panel discussion of Diesel Locomotive Crankcase Oil Filtration. Representatives of the railroads, the filter manufacturers, the engine manufacturers and the lubricant suppliers will be on this panel, and this virtually assures a most interesting, lively, and worthwhile session.

General Session

Some excellent papers which do not fit into any of the foregoing categories have been combined into a general session. These include papers on rust preventives, radiation resistant greases, fire resistant turbine fluids, lubricant disposal and marine hydraulic applications.

This, then, is a preview of the program, and it should be apparent that there is something here for everyone. You can't afford to stay away; so make plans *now* to attend. It must be emphasized that this is a full three day meeting, and the sessions to be held on the afternoon of the third day will be as interesting and informative as those that precede it. So come prepared to stay to the end.

ASLE Annual Meeting
April 11, 12, and 13, 1961
Bellevue-Stratford Hotel
Philadelphia, Pa.

The ASLE Banquet on April 12, 1961 will be the featured social event during the three days of technical sessions and activities of the 1961 Annual Meeting.
Banquet tickets will be available at the registration desk, so plan to get yours when you register. And remember, guests are invited.

SPECIAL EVENTS

April 10 – Philadelphia Section Welcoming Party 5:30 P.M.-7:00 P.M.— Clover Room, Bellevue Stratford

Ladies Program Features

Tuesday – Luncheon, Fashion and Fibre Show, Bellevue Stratford

Wednesday– Tour and Luncheon — Valley Forge

(Consult program for details)

For the Banquet Guest's enjoyment . . .

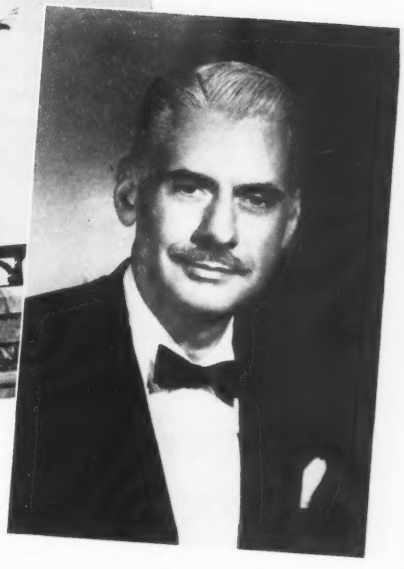
. . . the Philadelphia Host Section has assembled some excellent talent to provide entertainment.

Wayne Barrie, a famous Mutual Broadcasting artist, will act as master of ceremonies. His entertaining personality and limitless repertoire on the electronic accordion will insure audience response to a fast moving evening.

The delightfully different acrobatic dance team, called Ray Romaine and Claire, will bring new angles to the old routines and give some new ideas in the art of dancing.

The concluding act will feature the superb baritone performance of Wilbur Evans, the internationally famous musical comedy singing artist and star from South Pacific

Ray Romaine and Claire —delightfully different.



Wilbur Evans— international singing star.

Friction Characteristics of

AUTOMATIC TRANSMISSION FLUIDS

as Related to Transmission Operation

by Merrill L. Haviland and John J. Rodgers

Research Laboratories, General Motors Corporation, Warren, Michigan

Shift performance in automatic transmissions has been found to be related to the frictional qualities of transmission fluids. This relationship has been studied in two types of apparatus—one a full-scale transmission setup and the other a bench apparatus.

In full-scale transmissions, fluids were evaluated with respect to shift performance, severity of stick-slip, and oxidation resistance. Shift performance and severity of stick-slip were determined by analysis of recorded variables such as output shaft torque and engine speed as functions of time. By measuring fluid acidity periodically during each investigation, an interrelationship between fluid oxidation resistance and stick-slip was found.

In the bench apparatus, measurements of friction as a function of sliding speed were made with load and fluid temperature as parameters. These measurements were made not only with new fluids but also with fluids conditioned by transmission use. A correlation was found between fluid performance in a transmission and the friction characteristics determined in the bench apparatus. Based on the fluid friction characteristics, pertinent comments are made regarding the friction mechanism.

INTRODUCTION

Since the introduction of the automatic transmission, two problem areas associated with its operation have been fluid oxidation (sludge and varnish formation) and squawk (1). With the introduction of the controlled coupling automatic transmission in 1956 automobiles, these problem areas were accentuated. Investigations indicated that these problems were related to transmission fluid quality and were brought about by the mixing of air with the fluid at elevated temperatures. It was thought that the resulting fluid oxidation was the cause of clutch plate squawk.

In an attempt to better understand the friction characteristics of fluids and the relationship between oxidation and squawk, a study of fluid performance was conducted using a Transmission Cycling Test which is similar to the Durability Test used in the qualification of Automatic Transmission Fluid, Type A, Suffix A Identification (2).

Although squawk is known to be a friction phenomenon, it is difficult to evaluate the friction properties of fluids in the full-scale transmission. For this reason, the friction properties of the fluids were studied in a bench apparatus. A correlation is made between the friction properties of fluids and the performance of the fluids in the transmission.

APPARATUS AND PROCEDURES

Transmission Cycling Test

The full-scale apparatus used to determine fluid performance consisted of a 1958 controlled coupling automatic transmission connected to an inertia flywheel and dynamometer. The transmission was driven by a production engine. After a break-in period to condition the clutch plates, a transmission fluid was subjected to 100 hours of cycling operation during which time the fluid sump temperature was maintained at 275 $\pm$ 5F. Each cycle was 1 minute in duration and consisted of a nearly full-throttle engine acceleration from idle through the four transmission speed ranges followed by a closed-throttle deceleration to idle conditions. A schematic diagram of the test setup is shown in Fig. 1.

The parameters that regulated the transmission shifting operations were controlled and recorded by means of an oscillograph.

Presented at the 15th ASLE Annual Meeting, Cincinnati, Ohio, April, 1960

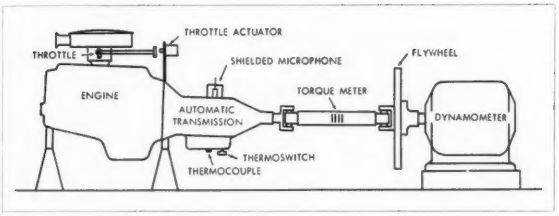


Fig. 1. Schematic Diagram of Test Setup.

Friction Apparatus

The Low Velocity Friction Apparatus was designed to simulate transmission clutch plate sliding. An upper steel specimen was rotated against a 1-inch mean diameter, 1/16-inch wide paper annulus (clutch plate material), in the presence of various lubricants. Friction was measured as a function of sliding speed, with load and fluid temperature as parameters. Both sliding surfaces were completely submerged in the lubricant. A schematic diagram of the essential components of the apparatus is shown in Fig. 2.

The friction between the rubbing surfaces was measured using a torque arm—strain gauge arrangement. In order to make accurate measurements of the low friction torque, external friction was minimized by floating the lower assembly in air. The normal force between the rubbing surfaces was varied by regulating the air pressure on the bottom of the floated assembly.

DISCUSSION

Transmission Shift Performance

Shift performance was evaluated in the transmission by observing output shaft torque and engine speed as functions of time. These variables, along with some of the parameters that govern shift characteristics, are shown in the oscillograph record in Fig. 3. In this figure the acceleration of the flywheel is indicated by the slope of the dynamometer speed trace. During the acceleration portion of the cycle, transmission throttle valve pressure, main line pressure, dynamometer load, and engine throttle position remain constant.

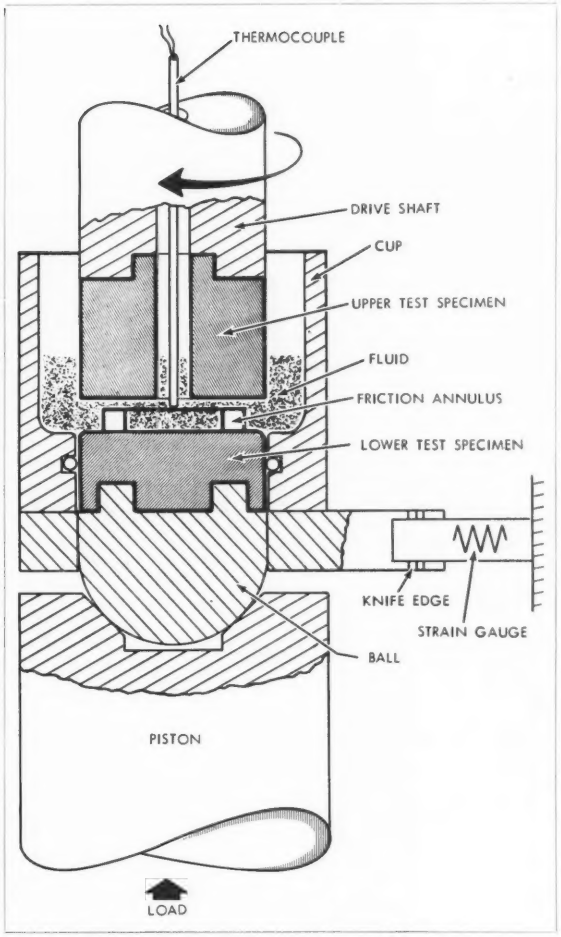


Fig. 2. Low Velocity Friction Apparatus.

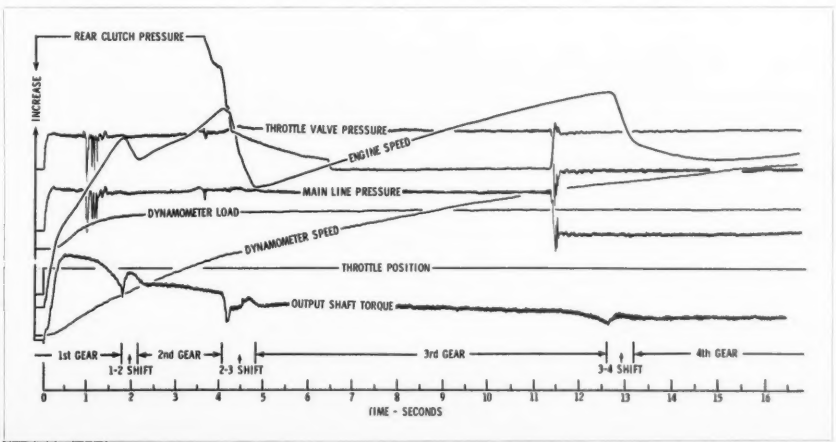


Fig. 3. Typical Oscillograph Record of Shift Characteristics and Transmission Variables.

Shift performance during the 2-3 shift is primarily dependent on the friction characteristics of fluids since the shift is accomplished by engagement of clutch plates. The other two forward shifts are accomplished by means of a fluid coupling and are inherently smooth. The 2-3 shift in Fig. 3 is representative of a smooth shift.

Fig. 4 illustrates the difference between satisfactory and unsatisfactory shift characteristics. With the unsatisfactory shift, the decrease in engine speed is nonlinear. In the figure shown, the time to complete the 2-3 shift has also been lengthened indicating undesirable clutch plate slippage. In addition, rapid fluctuations were observed in the output torque trace.

The torque fluctuations mentioned in relation to the unsatisfactory shift in Fig. 4 are the result of clutch plate action termed stick-slip. Stick-slip is a phenomenon in which the friction between the clutch plates increases with decreasing relative sliding speed until "stick" or adhesion of the surfaces occurs (3). When the clutch plates "stick," the torsional forces within the system increase (4). When the forces exceed the friction force, rapid slip occurs. This cycle may repeat itself several times during a single clutch engagement, and when the condition becomes severe, resonant vibration of transmission parts may occur resulting in an objectionable noise termed squawk. Moreover, severe stick-slip may cause fatigue failure of transmission parts.

Transmission Test Results

The performance of several automatic transmission fluids has been evaluated in the full-scale transmission test. In these evaluations, stick-slip and clutch plate slippage were the criteria used in determining fluid performance. Results from these tests and car tests are summarized in Table 1. In this table, information is presented for thirteen fluids, each containing one of seven different package additives. These fluids are commercial fluids and, therefore, the constituents in the fluids and additives are not known.

The package additives generally perform several functions. The effect on friction characteristics and the resistance to oxidation are the two functions most relevant to this investigation. Seven of the fluids (A, B, C, D, E, F, and M) which contain package additive A were judged satisfactory in car performance. Six of the fluids containing the additive A (Fuids A, B, C, D, E, and F) were tested in the Transmission Cycling Test. All six fluids were judged satisfactory, even though three of the fluids (B, C, and F) permitted initial stick-slip (present at the beginning of a test—becomes nonexistent as the test progresses) and another one (Fluid E) permitted stick-slip to begin just prior to completion of the 6000 cycle test. Two fluids (C and F) prevented the recurrence of stick-slip in the normal test duration. However, stick-slip did recur when these tests were extended considerably beyond 6000 cycles. Fuids G and H were not satisfactory in the transmis-

TABLE 1.—Fluid Composition and Summary of Fluid Performance

AUTOMATIC TRANSMISSION CYCLING TESTS							CAR TESTS
			SHIFTING PERFORMANCE				
			STICK-SLIP		CLUTCH PLATE SLIP (SHIFT TIME INCREASE—SECONDS)	FLUID OXIDATION (TAN INCREASE)	AUTOMATIC TRANSMISSION SHIFTING PERFORMANCE
FLUID	ADDITIVE TYPE	LENGTH OF TEST (CYCLES)	OCCURRENCE (CYCLES)	SEVERITY			
A	Pkg. A	6,000	None		−0.1	−0.53	Satisfactory
B	Pkg. A	6,000	0-600	Light	−0.1	+0.79	Satisfactory
C	Pkg. A	11,612	0-300	Light	0	+1.78	Satisfactory
			9000-11,550	Light			
			11,550-11,612	Light Squawk			
D	Pkg. A	8,037	None		0	−0.44	Satisfactory
E	Pkg. A	6,003	5700-6003	Light	+0.4	−0.84	Satisfactory
F	Pkg. A	7,052	0-600	Light	0	+0.10	Satisfactory
			6000-7052	Light			
G	Pkg. B	3,800	0-1200	Light	+1.2	+5.74 (Varnish)	Satisfactory
			1200-3800	Loud Squawk			
H	Pkg. C	2,598	0-2598	Loud Squawk	0	+3.13 (Sludge)	No Information
J	Pkg. D	No Information					Satisfactory
K	Pkg. E	No Information					Squawk (Worse as Temp. Increased)
L	Pkg. F	No Information					Squawk (Worse as Temp. Decreased)
M	Pkg. A	No Information					Satisfactory
N	Pkg. G	No Information					Satisfactory

sion test, because stick-slip persisted throughout each test.

From the foregoing information it can be seen that, depending on the fluids, stick-slip may be present at the beginning of a test; it may continue throughout a test; or it may be initiated within the duration of the test.

It is believed that initial stick-slip is the result of clutch plate surface conditioning. However, fluid friction properties are involved since initial stick-slip did not always occur. In any event, little significance is placed on intial stick-slip since no harm has resulted from it.

As mentioned previously, one important aspect of package additives is the effect on fluid friction characteristics. To illustrate this point the performance of fluids containing additives A, B, and C is compared. Fluids containing additive A were judged satisfactory in transmission operation, whereas fluids containing additives B or C permitted objectionable stick-slip throughout the test. Furthermore, stick-slip, indicated by the torque fluctuations, with additives B and C was so severe that squawk occurred.

The second criterion in shift performance, clutch plate slippage, is indicated in Table 1. With Fluid G, the clutch engagement time increased 1.2 seconds in about 3800 cycles, whereas no appreciable increase in engagement time was observed with the other fluids investigated.

The other additive effect remaining to be discussed is the contribution of additives to oxidation resistance. Certain additives provide greater protection against fluid oxidation than do others. Total acid number (TAN) analyses of Fluids C, G, and H from transmission tests demonstrate the differences in oxidation resistance of additives A, B, and C respectively. In Fig. 5 the TAN of these fluids is shown as the test progressed. With additives B or C (Fluids G and H) oxidation occurred at about 1000 cycles, as indicated by an increase in TAN. The TAN of the fluid with additive A (Fluid C) did not increase until about 11,000 cycles, indicating much greater oxidation resistance. Squawk was observed with all three fluids after the TAN had increased.

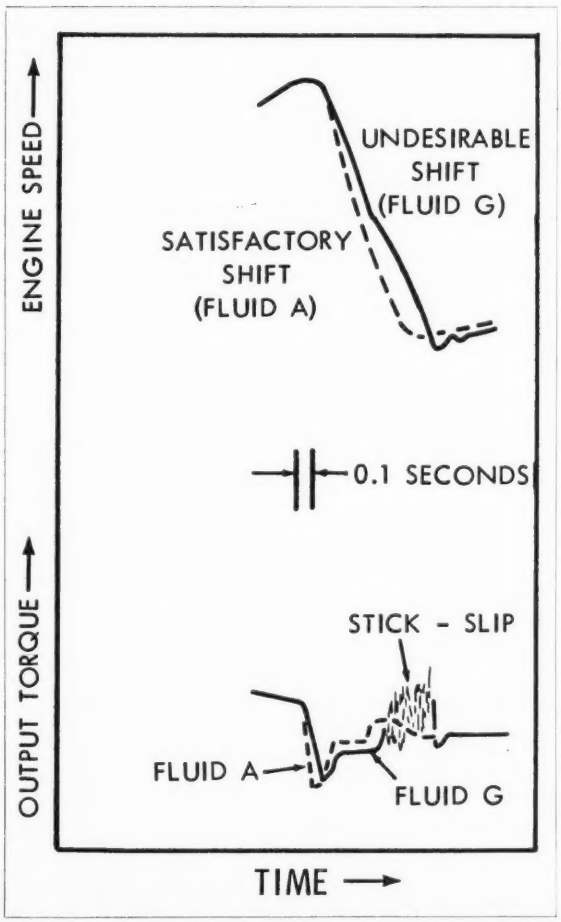


Fig. 4. Comparison of Satisfactory and Undesirable Shift Performance.

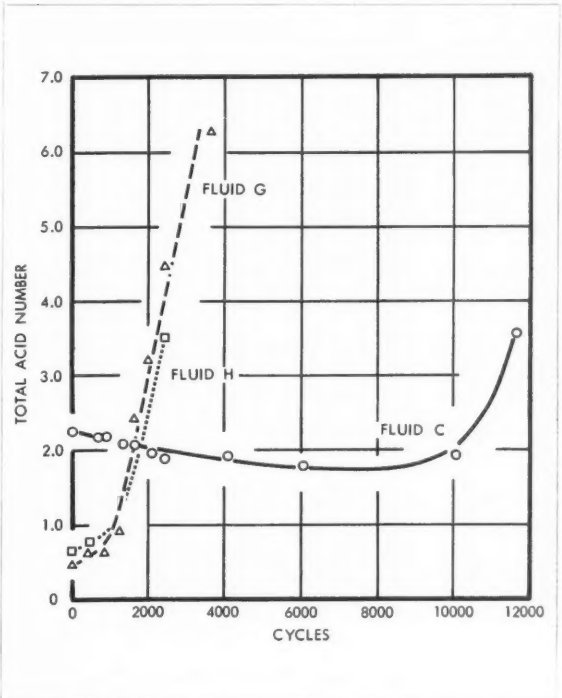


Fig. 5. Fluid Oxidation in Transmission Operation.

Friction Apparatus Results

Although the transmission test affords a means of evaluating fluid performance, it cannot be easily adapted to investigate the friction properties controlling shift performance. As a result, the friction apparatus was used to study the friction characteristics of fluids. Preliminary tests with this apparatus indicated that shift performance in the transmission could be explained on a frictional basis.

For example, clutch engagement patterns characteristic of both satisfactory and undesirable shift performance are shown in Fig. 6. The satisfactory shift was obtained with Fluid A whereas the unsatisfactory shift was obtained with Fluid H. The friction characteristics of the two fluids are presented in Fig. 7 where coefficient of friction is shown as a function of sliding speed. There are two obvious differences in these friction curves—the over-all shape, and the friction values at any one sliding speed. With Fluid A the coefficient of friction remains essentially constant above a sliding speed of 20 fpm, but below this speed the coefficient decreases as the sliding speed approaches zero. With Fluid H the coefficient of friction is also constant above 20 fpm; however, the value is approximately half that obtained with Fuid A. Furthermore, the coefficient of friction with Fluid H increases as the sliding speed is reduced below 20 fpm. On the basis of these fluid differences the shift patterns in Fig. 6 can be explained.

During the initial portion of the clutch plate engagement, the relative sliding speed of the clutch plates in the transmission is high. Since the coefficient of friction with Fluid A is greater than with Fluid H in the bench apparatus high sliding speed region (above 20 fpm), the initial change in sliding speed (clutch plate slippage) is greater with Fluid A than with Fluid H. On this basis, the shift time with Fluid A should be less than with Fluid H. However, once the sliding speed drops to lower values, the coefficient of friction with Fluid A decreases, whereas that with Fluid H increases.

The greater friction with Fluid H causes a rapid drop in the sliding speed, resulting in an abrupt shift. Such an abrupt shift is characteristic of stick-slip which was evidenced by rapid fluctuations in the torque trace. Thus, any fluids having friction characteristics similar to Fluid H would probably result in stick-slip.

From the foregoing discussion, it can be seen that fluid performance in a transmission can be predicted by the friction characteristics as determined in the bench apparatus. Fluids that exhibit an increase in the coefficient of friction with a decrease in sliding speed are undesirable. On the other hand, satisfactory fluids exhibit no such increase.

The bench apparatus was used to determine the effects of fluids, temperature, load, and transmission operation on friction characteristics.

Effect of Fluids: As mentioned earlier, initial stick-slip was frequently experienced in transmission operation. Similarly, friction characteristics that are indicative of stick-slip were generally obtained in the bench apparatus before break-in or conditioning of the surfaces occurred. However, during the break-in period (load—120 psi, sliding speed—40 fpm, time—14 hours) the friction characteristics of the surfaces changed until finally an equilibrium condition was reached where no further change was observed. All friction characteristics reported in this paper were obtained after break-in.

Friction results for several automatic transmission fluids at a temperature of 85 F are presented in Fig. 8. Comparing the friction characteristics of the fluids with stick-slip tendencies in transmissions (Table 1), the satisfactory fluids exhibited no increase

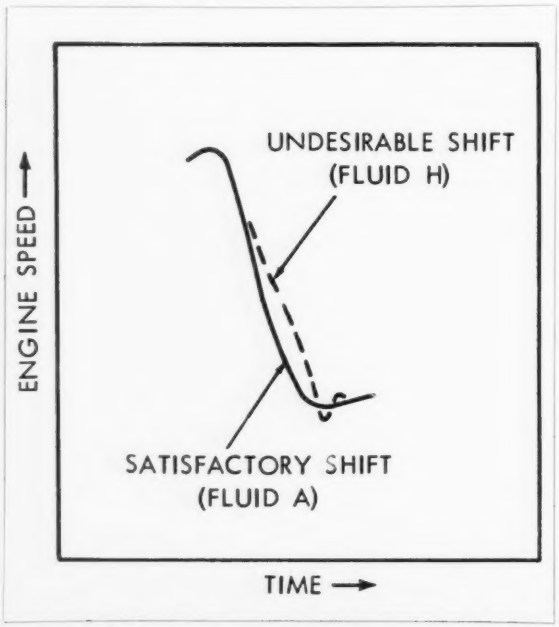


Fig. 6. Satisfactory and Undesirable Shift Patterns

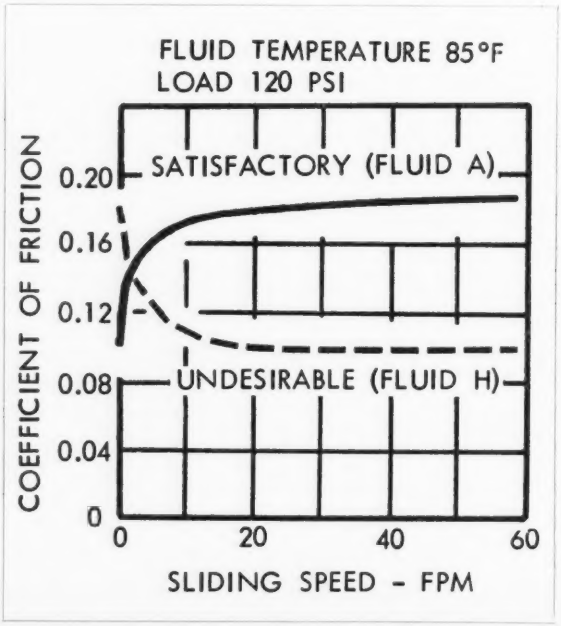


Fig. 7. Satisfactory and Undesirable Friction Characteristics

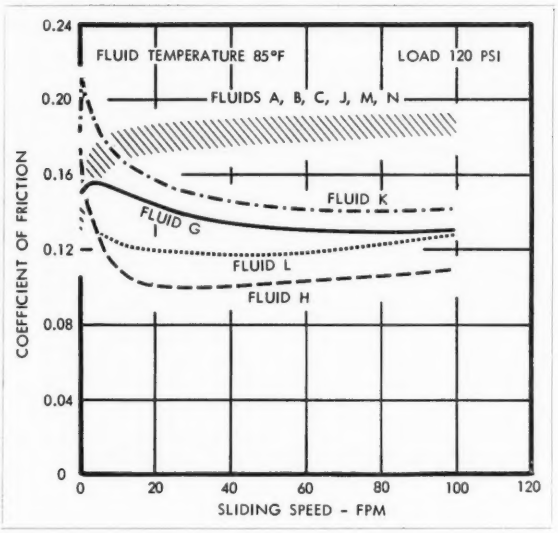


Fig. 8. Friction Characteristics of Several ATF's.

in coefficient of friction with decreasing sliding speed. Also, it is of interest that the satisfactory fluids exhibit very nearly the same characteristics as represented by the shaded area in the figure. On the other hand, the four fluids (G, H, K, and L) which produce stick-slip in the transmission exhibit unsatisfactory friction characteristics.

Effect of Temperature: The effect of temperature on the fluid friction characteristics is illustrated in Fig. 9. The results with all satisfactory fluids were almost identical and are represented by those of Fluid B. Increasing the temperature did not affect the characteristic shape of the friction curves. However, the coefficient of friction values at all sliding speeds including zero (static) decreased as the fluid temperature was increased from 85 to 300 F.

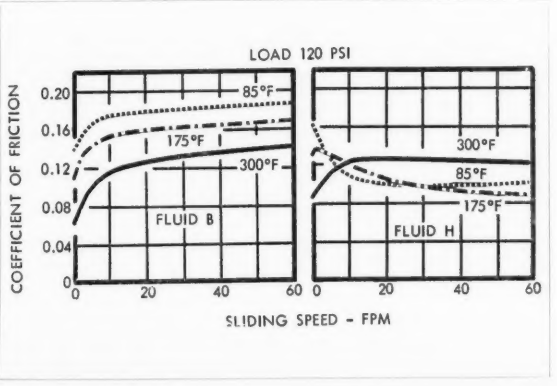


Fig. 9. Effect of Fluid Temperature on Friction Characteristics.

The effect of increasing fluid temperature on the friction characteristics of the undesirable fluids was a general improvement as represented by those for Fluid H. The fact that the friction characteristics become desirable with increasing fluid temperature suggests that stick-slip tendencies in a transmission would lessen; however, transmission experience indicates that the opposite is generally true. The reason for this discrepancy may be that clutch plate surface temperature in the transmission is much higher than the bulk fluid temperature. In the bench experiments, the surface temperature was essentially the same as the bulk fluid temperature.

However, the bench apparatus can be used to predict fluid performance as has been shown. Fluids exhibiting desirable friction characteristics throughout the temperature range (85 to 300 F) perform satisfactorily in transmissions. Fluids exhibiting undesirable friction characteristics at any temperature do not perform satisfactorily.

Effect of Load: The effect of three loads, 50, 120, and 180 psi, on the friction characteristics of the fluids was determined. The pressure effect was similar for all fluids. The general friction characteristics of the fluids remained unchanged as the load was increased. However, the values of static coefficients of friction remained essentially constant, whereas the values of kinetic coefficients of friction were reduced as the load was increased.

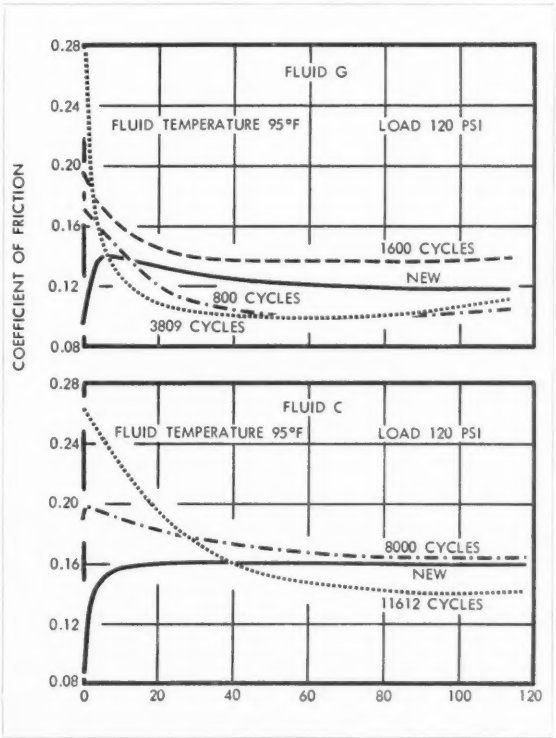


Fig. 10. Change in Friction Characteristics with Transmission Operation.

Effect of Transmission Operation: In the transmission tests discussed earlier, a marked increase in the TAN of fluids was observed at the occurrence of squawk. To show the change in fluid friction characteristics accompanying squawk and oxidation, fluid samples taken during the course of the transmission tests were studied in the bench apparatus. Results from two tests are shown in Fig. 10.

First consider the friction changes with an unsatisfactory fluid (G). Even at the beginning of the test where stick-slip was observed, this fluid exhibited undesirable friction characteristics as indicated by an increase in coefficient of friction with decreasing sliding speed. As the test progressed, stick-slip became more severe and resulted in squawk. At the same time the friction characteristics became increasingly undesirable.

Next consider friction changes with satisfactory Fluid C. Extending the test beyond 6000 cycles results in pronounced changes in friction characteristics. Eventually, the characteristics become definitely undesirable. For example, at 8000 cycles the friction characteristics are certainly indicative of stick-slip, although stick-slip in the transmission was not observed until about 9000 cycles and TAN did not increase until about 10,000 cycles.

On the basis of these analyses, it becomes apparent that friction characteristics of fluids change before fluid oxidation is indicated by a TAN increase or before stick-slip is observed in the transmission.

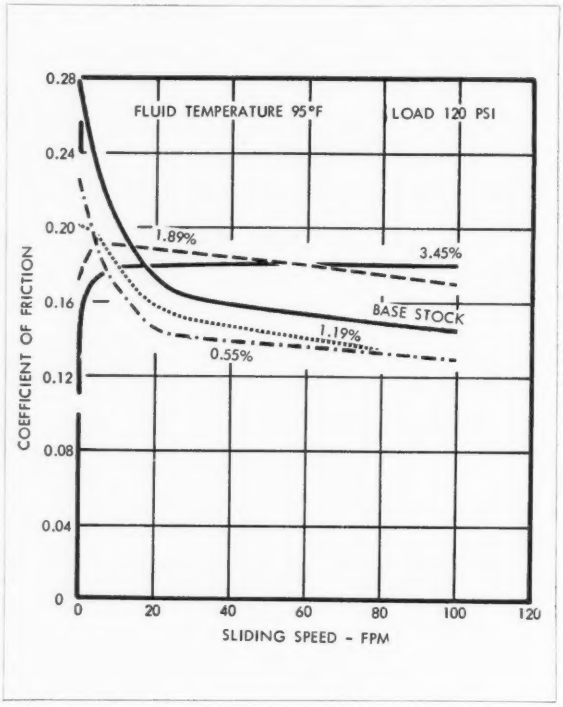


Fig. 11. Effect of Additive A Concentration on Friction Characteristics.

Friction Mechanism

In order to explain the friction mechanism associated with clutch plate sliding, a thorough knowledge of the physical and chemical properties of the sliding system is required. In this analysis no attempt has been made to describe the surface reactions that most likely take place. However, based on the friction characteristics and transmission performance of various fluids, some pertinent comments regarding the friction mechanism can be made.

Involved in the sliding system are three components—the steel surface, the paper surface, and the lubricant. Since, in this investigation, only one sliding combination was used (paper on steel), the differences in the friction characteristics observed are primarily due to the friction properties of the fluids.

Lubricants: The main differences in the friction characteristics of the lubricants studied are thought to be due to the particular combination of additive compounds, the ability of the various compounds to attach themselves to the surfaces, and the concentration of compounds present (5). To illustrate the concentration effect on the friction characteristics, friction measurements were made with various concentrations of Package Additive A in a base stock. These results are shown in Fig. 11. It is seen that the friction characteristics of the base stock and additive concentrations up to 1.19% are indicative of clutch plate stick-slip. Desirable friction characteristics were obtained when the Additive A concentration was increased to 1.89%. Above this concentration, there was essentially no change in the friction characteristics. Thus, an active additive compound will alter the friction characteristics of a base stock to permit satisfactory clutch plate operation only if the concentration of the additive is sufficient.

Friction Characteristics: There are two obvious differences that exist between satisfactory and undesirable friction characteristics: First, the distinct difference in slope of the friction curves at the low sliding speeds, and, second, the difference in the values of coefficient of friction, which cannot be attributed to hydrodynamic lubrication, at the higher sliding speeds.

It is thought that the desirable friction characteristics result from a reduction of the metal-to-fiber contact area. Such a reduction is attributed to the presence of additive molecules between the surfaces (3). These molecules are thought to be long-chain hydrocarbons which form a close-packed film on the surfaces (6, 7).

When friction characteristics are indicative of stick-slip, there is a large area of metal-to-fiber contact, particularly at low sliding speeds. The resulting friction is high since the paper fibers must be deformed to allow sliding. As the speed increases, less fiber deformation occurs because the oncoming metal asperity contacts the fiber before it has time to recover to its original position. Hence the friction decreases.

In the high sliding speed region, where desirable fluids exhibit a higher coefficient of friction than do the undesirable fluids, it is thought that the increase in friction does not result from fiber deformation. It is suggested that the difference in friction force is due to molecular shearing and abrasion of the molecules from the surfaces (8, 9, 10).

The number of molecules abraded and sheared is reduced with decreasing sliding speed. In addition, there is more time for repair of the depleted lubricant film (5). As a result, the coefficient of friction with satisfactory fluids decreases with decreasing sliding speed.

SUMMARY

Automatic transmission fluids have been evaluated for shift performance in a full-scale transmission cycling test. In addition, the friction characteristics of many of these fluids were determined in a friction apparatus. Results from these studies showed a definite correlation between the friction characteristics and the shift performance of fluids. It was found that fluids providing smooth clutch engagements exhibited no increase in coefficient of friction with decreasing sliding speed. On the other hand, undesirable fluids produced abrupt shifts, stick-slip, and in some cases squawk. In these cases the coefficient of friction was found to increase with decreasing sliding speed. Moreover, with satisfactory fluids the kinetic coefficient of friction was higher than that with undesirable fluids.

Friction characteristics as affected by load, temperature, and fluid conditioning in the transmission have also been determined. It was found that load did not affect the general friction characteristics of fluids, although an increase in load did reduce the values of kinetic coefficient of friction.

An increase in the temperature of satisfactory fluids caused a reduction in friction values throughout the sliding speed range of the bench apparatus. The effect of increasing the temperature of undesirable fluids was a general improvement in friction characteristics. On the other hand, transmission operation resulted in a definite depreciation of the friction characteristics for all fluids evaluated. This depreciation occurred before fluid oxidation was observed. In the transmission, oxidized fluids produced squawk; however, fluid oxidation was not necessary to produce stick-slip.

ACKNOWLEDGMENT

The authors wish to express appreciation to members of the Fuels and Lubricants Department for their comments during the preparation of the manuscript. Particular thanks go to N. A. Hunstad for his support in the area of study and to D. R. Lawson for the basic design of the friction apparatus.

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What Price Lubrication

by Allen F. Brewer,

Consultant and Author, "Basic Lubrication Practice"

Management has a very definite stake in effective lubrication. Costwise as compared with general operating expenses the price paid for lubrication usually is not alarming. Insurancewise, however, as a protective measure for plant investment the value is inestimable. The cost of lubrication can be determined most significantly from records. There are of course certain contingencies or factors which require coordination between management and the lubrication engineer. Evaluation of the operating conditions with respect to lubricants and lubricating systems is a basic requirement. But equally as essential are records of lubricant consumption and performance. When these data are assembled in tabular form the prospective benefits to management will include increased machine life, more dependable production, lower operating and maintenance costs, and improved safety conditions. When management realizes that the lubrication engineer is most capable of coordinating the lubrication requirements of their plant machinery with available lubricants and lubricating equipment, the ratio of cost of lubrication to the cost of production should be agreeably low. "What Price Lubrication" outlines the procedures to attain this objective.

The relation of price to lubrication must be considered with respect to certain influencing factors which can have a decided effect upon the ultimate cost. Since management is concerned with cost these factors can become the basis upon which the lubrication engineer can develop his personal status as a respected member of the plant managerial family. Viz.;

Evaluation—or study of the operating conditions with respect to the physical and chemical characteristics of available lubricants.

Significance—of the physical and chemical characteristics of the lubricants with respect to these operating conditions.

Statistics—or recording of lubricant consumption and performance.

Coordination—of lubricating personnel with the operating and maintenance people.

THE BENEFITS TO MANAGEMENT

The payoff is significant because it definitely proves the importance of the lubrication engineer and his contribution to reduced operating costs and lower maintenance expense. Management benefits through

Increased Machine Life, because premature wear-out is prevented when lubricants are designed to function in the lubricating systems and chemically adapted to the service conditions.

More Pay Load. Effective lubrication insures greater pay load. Machinery can be operated more continuously and dependably at over-speed and overload conditions when film strength is resistant to the squeeze-out, throw-off effects of such service. Film strength is obtained by scientific study of compatibility of the base lubricant with the most suitable EP additive. It cannot be assured by "price buying".

Lower Operating Costs result when a sound, well-built machine is provided with component parts designed for perfectly synchronized motion and effective lubrication. When these parts can function continuously so that there is no over-heating, no abnormal power consumption, no unusual shut-down for adjustment, maintenance or repair, the unit operating cost of production should be low.

Greater Safety. When machinery is designed for safe operation everybody profits. It hurts to be hurt. In a machine this may simply involve increased repair or maintenance costs. With plant personnel, however, unsafe machinery or hazardous operating conditions could mean injury, hospitalization, loss of time and income. To the management it could mean increased insurance rates and possibly costly litigation. Effective lubrication plays a most important part in assuring that modern machinery is operating for greatest safety.

Reduced Maintenance Costs. Low cost maintenance means low cost production; normally there is less time out for repair and less production loss. The lubrication engineer can play an active part in keeping maintenance costs down when he has developed records of such costs and can balance them against cost of production and the unit cost of lubrication. See Figs. 2 and 3 which show examples of forms that may be used to facilitate record keeping.

Presented at the ASLE Boston Section Meeting, March 14, 1960.

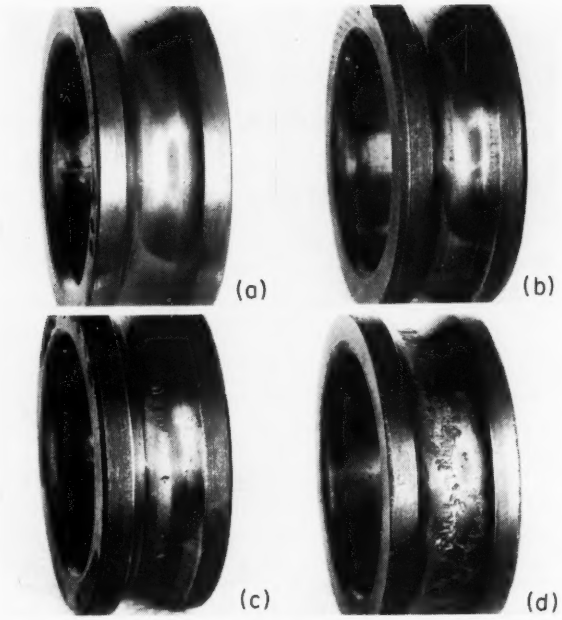

Fig. 1. Progressive stages of flaking in a deep groove ball bearing race caused by insufficient lubrication. (a) shows the initial stage with (b), (c) and (d) showing subsequent stages of the damage. Taken from the ASLE publication *Interpreting Service Damage in Rolling Type Bearings.*

LUBRICANT CONSUMPTION RECORD Month of _____

LOCATION, NAME OF MACHINE OR TYPE OF MECHANISM						
Grade of Lubricant	Date Received	Amount Received (A)	Amount on hand, end of current month (B)	Volume consumed for current month (C)	Machine hours operated during current month (D)	Unit lubricant consumption
oil or grease		gallons or pounds charged out from oil house or storage	from inventory	(A)-(B)	Obtained from operating records	(C) / (D)

Fig. 2. Lubrication Consumption Record Form.

MONTHLY LUBRICATION SUMMARY Month of _____

Grade of Lubricant	Volume consumed for current month (A)	Unit cost per gallon or pound (B)	Total cost		Percent	
			Current month (C)	Previous month (D)	increase	decrease
From Table 1	Column (C) Table 1	Taken from invoices	(A)x(B)		Column (C) compared with column (D)	Column (C) compared with column (D)

Fig. 3. Monthly Lubrication Summary Form.

TOTAL MONTHLY COST OF PLANT LUBRICATION Month of_____

Cost of lubricants consumed (from Table 2, Column C)	
Labor charged to lubrication department	
Cost of supervision, (allocated if necessary)	
Maintenance of lubricating equipment	
New lubricating equipment installed, or carried as inventory	
Storage charges	
Total	$

Fig. 4. Total Monthly Cost of Plant Lubrication Form.

LUBRICATION RECORD

Machine identity, with listing of operating parts	Type of lubrication	Lubricant, type or grade	Re-lubrication periods	Frequency of inspection
Bearings Sleeve-type Ball Roller				
Gears Enclosed Exposed				
Chain drives Enclosed Exposed				
Wire rope				
Couplings lubricated flexible				
Cams				

Fig. 5. Lubrication Record Form.

PROFIT FROM RECORDS

This requires complete cooperation between management and the lubricating personnel. Records can be kept in a variety of ways. In reality their presentation can reflect the personality of the lubricating people as well as the machine. Simplification is conducive to accuracy, readability and associating the cost of lubrication with the cost of production. Then manufacturing costs as related to lubrication become more realistic. The forms shown in Figs. 2 and 3, along with Fig. 4 are of value when calculating the total monthly cost of plant lubrication. Forms similar to those shown in Figs. 5 and 6 include other pertinent data which are of value in simplifying the lubrication program. Quite naturally simplified lubrication with reduction in the number of lubricants required, reduces the storage and inventory problems and the possibility of misusage of lubricants.

PERTINENT DESIGN AND OPERATING DETAILS

Identity of Machine _____

BEARINGS

Sleeve-type

Bearing metal	Operating temp. range °F.	Type of lubrication	Type of lubricant	Viscosity range or consistency

Ball or Roller

Design and type	Operating temp. range °F.	Type of lubrication	Type of lubricant	Viscosity range or consistency

REDUCTION GEARS AND CHAIN DRIVES

Type of housing	Operating temp. range °F.	Type of lubrication	Type of lubricant	Viscosity range

Fig. 6. Pertinent Design and Operating Details Form. This same scheme of data presentation can be followed for flexible couplings, power transmission units and geared motor drives.

THE PRICE RELATIONSHIP

The price of any finished product will include

(a) Cost of the raw materials.
(b) Cost of transportation to the plant for processing to finished product.
(c) Cost of labor for such processing.
(d) Time value of the essential machinery.
(e) Cost of handling in the plant.
(f) Cost of packaging for shipment.
(g) Sales and advertising costs.

All this applies to lubricants as well as to other material goods. On the other hand, when the market is highly competitive the background cost of lubricant production may be lost sight of by the purchaser. The tendency may be to match cents per gallon or per pound instead of considering prospective lubricant performance as the primary objective.

Performance involves much more than simply the ability of an oil or grease to function as a lubricant. It includes

(a) Stability of the lubricant when subjected to the prospective operating conditions.
(b) Its protective ability when in service in the machine.
(c) Its reliability when used in long-time service as, for example, in prepacked ball or roller bearings.

These are details with which management should be vitally concerned because a few more dollars spent for their assurance can mean operating insurance for machinery which can be worth several thousand times as much. Where lubricants are involved the purchaser is buying more than simple commodities. In addition to low cost maintenance and safety of plant personnel which have already been referred to as Management Benefits, he is buying

(a) Dependable machine operation, and
(b) Insurance against fire.

The lubrication engineer must make sure that these factors are not overlooked.

FORCED SHUTDOWN

Wear is bound to occur in certain parts of any mechanism; it can become especially severe where lubrication has been inadequate. The term shutdown, of course, must be considered according to the cause. There are the usual routine shutdowns of a machine, or possibly the entire plant, for adjustment, inspection and cleaning. There are also the periodic machine shutdowns for lubrication. The forced unexpected shutdowns are of most concern to management. When an unexpected shutdown is necessary due to faulty lubrication it may be on account of

(a) Using a non-effective lubricant.
(b) Using an excess of a suitable lubricant, which would result in leakage and contamination of the product being processed.
(c) Using insufficient lubricant to result in "starved" lubrication.

The wear and tear demands upon production or power plant machinery today are becoming more and more severe as units are speeded up and often overloaded to meet production or peak load schedules. Fortunately this is anticipated by

(a) The machinery builders.
(b) The Petroleum and Chemical Industries.
(c) The operating department and the lubrication engineer.

Machinery Builders design to meet those requirements by devoting the utmost attention to uniformity in metal hardness, closer tolerances, and utilizing modern methods of automatic application of lubricants.

The Petroleum and Chemical Industries study the prospective intensity of the operating conditions and the nature of the metals involved. Then they produce lubricating oils, greases and synthetic materials containing suitable additives to withstand load and to retard the tendency towards oxidation, rusting and corrosion.

When the *Lubrication Engineer* with the cooperation of the *Operating Department* can plan to use plant machinery as a proving ground to determine the suitability of any proposed lubricant for the prospective service, invaluable data often can be developed. These data can be used by all concerned in planning for applications of automatic lubricating systems best suited to the design and to the service conditions as a whole.

Past history must guide the designing engineer when incorporating any means of oil, grease or solid lubrication in his plans. Past history also is most helpful to the plant people when purchasing and applying the lubricants required by the operating conditions and the lubricating systems. A positive system designed for reasonable control of lubricant passage to the operating parts reduces the possibility of forced shutdown. Forms similar to those shown in Fig. 7 will be useful in making a fairly accurate calculation of the probable cost of shutdown.

ITEMS INVOLVED WHEN SHUTDOWN IS NECESSARY

Description	How Operation or Costs Are Affected	Time Loss Hours	Estimated or Calculated Cost
Production	In terms of output		
Plant procedure	Adverse affect on related plant machinery. (slowdown or actual stoppage)		
Repair and maintenance costs	Cost of parts renewed. Cost of labor for parts renewal		
Labor	Operators for idle machines must be released or detailed to other duties		
Specialty service	Expert labor or supervision required (beyond plant facilities) to get things running again		
Overhead	Must go on as usual even though production is curtailed		
Utilities	Must be maintained as usual, even though production is cut or curtailed		

Fig. 7. Form used to list items involved when shutdown is necessary.

ORGANIZATION AND RESPONSIBILITY

Responsibility for the organization and operation of the lubrication program plays an important part in cost control. It has a definite price relationship. Organization is a matter of teamwork. When the lubrication personnel "think lubrication" their value to management is obvious.

Background of course is important. An analytical viewpoint is essential. The lubrication engineer may have developed this naturally through active interest in practical mechanics; then realization of the value of lubrication becomes second nature. Or he may be a graduate engineer or chemist who has come up through the engineering or maintenance departments, sometimes having served as plant master mechanic or millwright supervisor. In any case when he assumes the duties of plant lubrication engineer and proceeds to organize, he should be ready to make the final decisions as to types and grades of lubricants required for the machinery involved. It also becomes his responsibility to impress upon his management that modern machinery is "designed for lubrication" as well as for production, by lubricants within definite viscosity or consistency ranges. He can be most convincing when he presents listing of all other pertinent physical and chemical characteristics which are desirable for effective lubrication under the prospective operating conditions.

Education and training are other important facets of the organization procedure. In service the lubrication engineer should be prepared to instruct machine operators as to the reasons for effective lubrication. He also must plan courses of study covering details of design, equipment limitations, lubricant characteristics and the part played by additives. He also should stress the importance of watching operating conditions for sudden increases in temperature, speed or load.

Lubrication schedules are highly important. With modern methods of automatic lubrication application procedure is largely mechanical; even so there must be supervision to make sure everything is on the beam. Other lubricating devices may require timing of the re-lubrication procedures and cleaning or flushing of systems periodically.

Routine inventory of stocks of lubricants on hand, for record purposes is another chore the lubrication engineer must assume, along with supervision of storage and distribution practices, to see that cleanliness is observed to prevent possible contamination. The storage problem is simplified when simplified lubrication studies have been made to reduce the number of types or grades required. The objective from the time any lubricant is packaged until it is charged into the lubricating system is *protection.* This involves the machine, the man, the product and the price of lubrication.

THE VALUE OF TIMING

Timed lubrication dates back to the days of the "tallow pot". Then it was manual and so often based on sound or feel. Modern procedure with completely automatic lubrication is to build timing right into the system. Then it has a definite relationship to price of lubrication. Waste is reduced, also leakage, and premium grade lubricants can be used most economically with greatest safety and reduced maintenance costs. In such systems the designers provide for delivery of oil or grease in adequate amounts to avoid any possibility of starved lubrication.

Timing and the lubrication schedule go hand-in-hand. The latter requires study of the machine: just how well its operating parts are housed or protected to enable retention of lubricant, the prevailing operating speed and temperature ranges, the types of seals and their ability to prevent lubricant contamination, and to what extent an oil can serve as a flushing medium. Records of plant production, maintenance costs and lubricant consumption are essential to working out an

effectual lubrication and timing schedule. Along with such records operating experience is invaluable, as well as thorough knowledge of the potential lubricating abilities of all the types or grades of lubricants being considered.

The potential value of timed lubrication as compared with manual handling of lubricating equipment, can be developed by recording time requirements and volume of lubricants (on a monthly basis) using a form similiar to that illustrated in Fig. 8.

TIMING DATA

	Method of Lubrication	
	By Manual Application	By Automatic Timed Application
Man hours for lubrication		*
Down time for maintenance or repair		
Volume of lubricants required monthly, (from Table 1)		

* Filling lubricators

Fig. 8. Timing data form.

MAINTENANCE OF LUBRICATING EQUIPMENT

The responsibility for maintenance of lubricating equipment may devolve upon the lubrication engineer or the maintenance department, according to the plant organization. In any case it covers the same objectives:

(a) Supervision of location and condition of all lubricating devices with appurtenant piping and fittings.
(b) Checking for leaks and the protective ability of seals and housings.
(c) Correction of functional deficiencies in any part of the system.
(d) Removal, cleaning and replacement of bearings, couplings, gears, chains and other lubricated parts.
(e) Cooperation with other departments sharing responsibility for repair of operating parts or protective housings.
(f) Checking on performance and cleanliness of oil filters, centrifuges, coolers and heaters.

Maintenance personnel should be machinists or millwrights who are thoroughly familiar with the entire assembly of machinery and the constructional features of the lubricating equipment. When they also are informed as to the characteristics of the lubricants being used they can more readily anticipate lubrication troubles before serious damage is done, by checking up on the cause. Their ability to judge the condition of bearing materials or to study gear tooth surfaces to determine whether wear or fatigue is the cause of the trouble, is a most valuable asset. When the cause, or any deficiency of the lubricating system is corrected in time maintenance becomes more than a remedy. It is preventive insurance against down time and loss of production.

Repair of lubricating equipment is an important phase of the maintenance program. Normally there will be a certain amount of piping or projecting parts which could be crushed, broken or damaged by careless handling of tools or contact with swinging crane chains, etc. If damage of this nature is not corrected immediately lubrication of the related machine parts could be seriously reduced. Subsequent wear of bearings or gear teeth could then develop into a major maintenance project, with attendant loss of production during down time for repairs.

A training program which instructs all plant personnel as to the importance of care in materials handling, and respect for lubricating equipment, should be the responsibility of the lubrication engineer. When he can illustrate by cost records of parts renewal or actual examples of damaged lubricating equipment, the program becomes all the more vivid especially if the safety factor is emphasized at the same time. Personnel so trained will find that it is second nature to make sure that lubricating equipment in their areas is in good working order.

Standardization is most important with respect to the maintenance program. It is inadvisable to have a wide variety of devices with their necessary fittings and spare parts. This can defeat the objective of low cost maintenance. Standardization will be logical when the lubrication department is consulted by management when systems are being considered. The lubrication engineer is most capable of evaluating the lubricating requirements and the ability of prospective equipment to meet these requirements with the least amount of maintenance.

Measuring the Oil Film Thickness in a Crankshaft

MAIN BEARING OF A V-8 ENGINE

by W. D. Sims,
Shell Development Company,
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An experiment has been conducted whereby the thickness of the oil film on the loaded side of the crankshaft bearing of an otherwise standard automotive engine was measured. Observations were made over a wide power range with oils of three widely differing viscosities.

The observations indicated that the oil film thickness between the crankshaft and the loaded side of the bearing varied from 0.002 inch at light load to approximately 0.0005 inch at full load and was essentially unaffected by the different viscosity oils.

The instrumentation employed in the measurements of bearing oil film thickness (clearance on the loaded side of bearing) in an operating V-8 automotive engine is described and the significance of the measurements obtained is discussed.

INTRODUCTION

The use of polymeric Viscosity Index improvers in engine lubricating oils entails the thickening of relatively thin (low viscosity) petroleum oils by small percentages of oil-soluble polymer. Such thickened blends have been shown to produce less engine friction, particularly at low temperatures, than wholly petroleum oils of comparable viscosity. This is a consequence of the non-Newtonian behavior of the polymeric-additive oils which makes them less viscous at the shear rates encountered in engines. Because of this lower viscosity at high shear locations, fears regarding the ability of such oils to adequately protect engine bearings, particularly after the bulk viscosity is reduced by the cumulative effects of fuel dilution and partial degradation of the polymeric additive during extended engine operation, have often been expressed. However, no case of bearing damage resulting from the use of polymer-containing oils in place of petroleum oils has been documented. Likewise, no data directly refuting such a possibility have been published. Furthermore, a perusal of the literature has led to the conclusion that it is impossible to predict, by calculation, the effect of lubricant viscosity on bearing performance in an actual engine. The inherent difficulties of measuring such parameters as individual bearing oil flow, oil supply pressure, oil film temperature, bearing loading, operating clearance and journal-to-bearing alignment lead to the employment of such gross assumptions as to render calculated values highly suspect.

In order to accurately assess the bearing lubricating ability of polymer-thickened oils, as opposed to oils blended from conventional lubricant stocks, it therefore was deemed necessary to devise means of measuring bearing oil film thickness in a nominally typical engine during actual operation obtained using oils representative of new and used polymeric-additive blends as well as petroleum oil of normally specified viscosity.

INSTRUMENTATION AND EXPERIMENTAL TECHNIQUE

The instrument selected for measurement of the thickness of a bearing oil film is based on the magnetic properties of a high frequency A-C sensing element. The sensing element is mounted in the end of a ½ inch x 20 thread capscrew. It is supplied with a 40 kc signal by a specifically matched oscillator-amplifier, the complete assembly forming a "dynamic micrometer". The output signal, which is displayed on a cathode-ray oscilloscope, varies with the proximity of a magnetic surface to the sensing element face. The response is approximately linear from zero to 0.030 inches. Previous experience has shown the response to be unaffected by the presence of engine oil in the magnetic clearance.

Presented at 15th ASLE Annual Meeting, Cincinnati, Ohio, April, 1960

Fig. 1. Main Bearing Cap and Sensing Element.

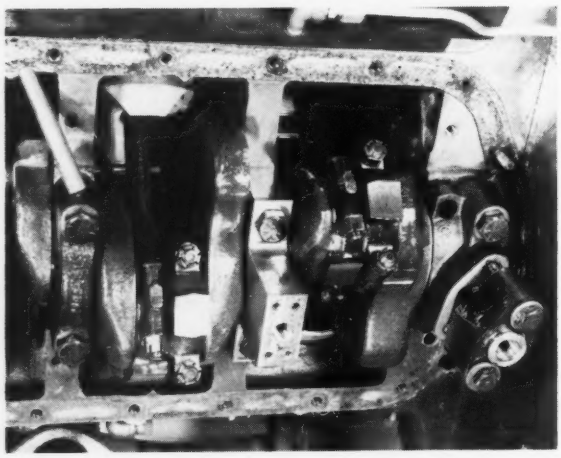

Fig. 2. Main Bearing Cap in No. 4 Bearing Position.

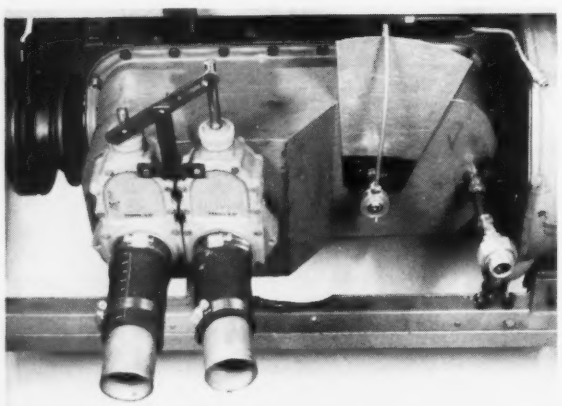

Fig. 3. Calibration Quadrant and Modified Oil Pan.

TABLE 1
TEST ENGINE SPECIFICATIONS

Engine Type	V-8, OHV
Total Displacement	331 cu. in.
Compression Ratio	9.0:1
Firing Order	1-8-4-3-6-5-7-2
Spark Advance	
Centrifugal	23.5° BTC/900-4000 rpm
Vacuum	27.5° BTC
Main Bearing Dimensions (except rear bearing)	2½ in. Dia. x 15/16 in.
Main Bearing Clearance (nominal, room temperature)	0.003 in.
Main Bearing Material	Steel-Backed Aluminum with Lead Overplate

The sensing element was installed in a modified main bearing cap of a modern V-8 engine. Engine and test bearing specifications will be found in Table 1 and the test bearing is shown in Figure 1. The ½-inch hole cut through the bearing insert for passage of the sensing element is also visible. Fig. 2 shows the bearing cap and pickup installed in the No. 4 position in the test engine. In Fig. 3 may be seen the quadrant and long arm employed to position the pickup, using the mounting thread as a micrometer screw. A 1-inch arc on the quadrant represented 0.001 inches vertical displacement of the pickup face. The two large slide valves on the crankcase (visible at the left) were part of the external oil system devised to perform oil changes on the run. This procedure allowed comparison of the operation on oils of varying viscosity to be made without long warm-up periods to obtain general temperature equilibria throughout the engine structure. An external centrifugal oil pump was employed to provide constant oil pressure (30 psi) unaffected by engine speed, etc.

The dynamic micrometer was calibrated by touching the face of the sensing element to the static crankshaft and setting the amplifier and oscilloscope gain to the desired pattern height (nominally 8 cm total deflection). Retaining these settings, the sensing suit was progressively moved away from the journal producing the calibration curve in Fig. 4.

After starting the engine, the pickup was returned to the "touching" position, i.e., flush with the ID of the bearing. The oil film produced by shaft rotation prevented actual contact during engine operation. A display of the typical oscilloscope pattern is shown in Fig. 5. Points of maximum and minimum clearance between the sensing unit face at the bottom of the bearing and the journal are indicated by arrows. The repeating deep grooves in the pattern are due to the passage of crankshaft oil holes past the sensing element. The pattern shown here was obtained at 2000 rpm, near full-throttle power output, using SAE 30 oil. Earlier runs made with the pickup installed in the No. 2 bearing position showed that varying speed from 1500 to 3600 rpm affected maximum and minimum clearance only slightly. For example, maximum clearance at both 2000 and 3000 rpm, full throttle, was 0.0041 inches. Minimum clearances observed at these speeds were 0.0003 and 0.0004 inches, respectively. Small changes in pattern were observed over the speed range indicating some response of the crankshaft to

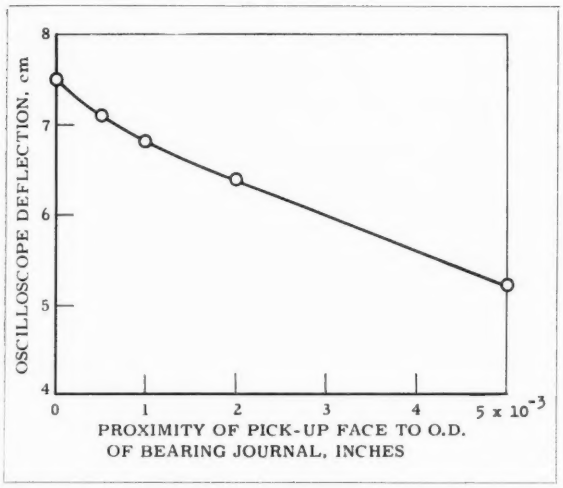

Fig. 4. Dynamic Micrometer Calibration.

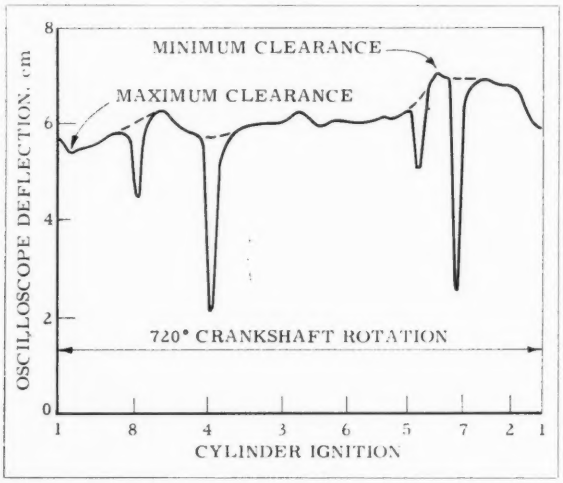

Fig. 5. Typical Oscilloscope Pattern.

frequency of the loading cycle or inherent minor unbalance. The magnitude of the speed-induced changes was, however, much smaller than those effected by power level. Hence, detailed study of the effects of power level and oil viscosity was confined to a single speed, 2000 rpm.

DISCUSSION OF RESULTS

The typical oscilloscope pattern of Fig. 5 is replotted as actual clearance vs. crankangle and engine firing order in Fig. 6. It will be noted that greatest clearance is obtained during the power stroke of No. 1 cylinder, which is located at the greatest possible distance from the test bearing. The minimum clearance occurs during the power stroke of No. 5 cylinder which imparts its load to the crankpin immediately forward of the test bearing. The downward force resulting in the minimum clearance is continued through the power stroke of No. 7 cylinder which is coupled to the crankthrow immediately behind the test bearing and fires only 90° after No. 5 cylinder. The firing of cylinder Nos. 8 and 3 may also be seen to force the journal downward in the test bearing while Nos. 4 and 6 cylinders appear to have little effect.

The distance between the journal and the sensing unit is shown in Fig. 6 to vary between 0.0005 and 0.0045 inches, a total travel in the vertical plane of 0.004 inches. As such a travel would be impossible within the 0.003 diametral clearance of the bearing at assembly, it is apparent that the actual running clearance of the bearing increased as the engine reached thermal equilibrium. Assuming the journal excursion is equal above and below the center of the bearing, i.e., approaching the same minimum oil film thickness of approximately 0.0005 inches, an actual running clearance of 0.005 inches is indicated.

The combustion events of the cylinders most directly affecting the test bearing are discernable in the clearance measurements. This observation is simplified by replotting the data as shown in Fig. 7. The events occurring in No. 8 cylinder are identified in the figure. The timing of the journal deflection vs. shaft rotation and firing order appear to be in agreement with the journal load data presented by Bidwell (1).

The effect of engine power level (BMEP) on the clearance between the sensing unit face and the test bearing journal is immediately apparent in Fig. 8. An increase in gas load on the pistons from 15 psi to 90 psi is shown to decrease the minimum clearance (apparent minimum oil film thickness) from 0.002 inches to 0.0005 inches.

As the instrumentation has been shown to adequately distinguish the cyclic events affecting load on the test bearing and to indicate response in a linear manner to engine loading, it appears adequate to reveal changes in bearing clearance, and hence in film thickness, caused by changes in other variables. Thus, any changes in film thickness due to bulk viscosity differences or to differences in viscosity behavior under high shear gradients should be readily observed. Results obtained with an oil comparable to SAE 30 petroleum oils (Oil A), a petroleum base component less viscous than those usually employed in polymeric-additive blends representing minimum viscosity possible with used polymeric oils (Oil B), and the latter thickened by the addition of 7.5%w polymethacrylate VI improver (Oil C), are also shown in Fig. 8. It will be noted that the clearance observations with all three oils, covering a viscosity range of 4.6 to 15 cs at the test temperature (see Table 2), were substantially identical at all power levels, indicating the viscosity of the oil to have no significant effect on clearance.

The glass bearing experiments of Cole and Hughes (2) have shown the presence of continuous oil films only on the loaded side of journal bearings, with only filaments of oil being present on the unloaded portion. The extension of their approach to include cyclic loading as described by Ozdas (3) showed the film to alternate so as to remain always on the loaded side of the journal. From these results it appears reasonable that the minimum clearance indicated between the sensing element and the bearing journal in the actual engine represents the thickness of the continuous oil film. The apparent absence of a viscosity effect must result from

TABLE 2
TEST OIL VISCOSITY DATA

OIL	Viscosity, Cs[1]				
	100 F	210 F	17″	10″	4″
Mineral Oil A	110.0	10.74	15.0	12.6	11.4
Mineral Oil B	25.7	4.56	6.5, 7.9	5.0, 6.9	4.6, 4.9
Mineral Oil B + 7.5%v polymethacrylate VI improver	38.0	7.92	10.9	9.0	8.2

1. Viscosity at 210° F determined on samples taken from test engine during operation. Viscosities shown at 17″, 10″ and 4″ Hg manifold vacuum calculated from viscosity at 100 F and 210 F and temperature of oil being supplied to bearing during dynamic micrometer observation via ASTM Method 341-43 Chart D.

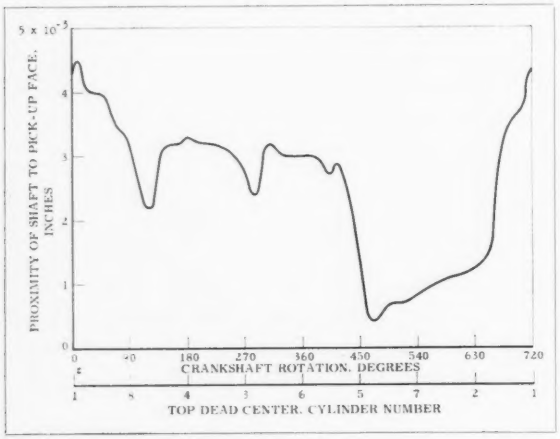

Fig. 6. Distance vs. Rotation.

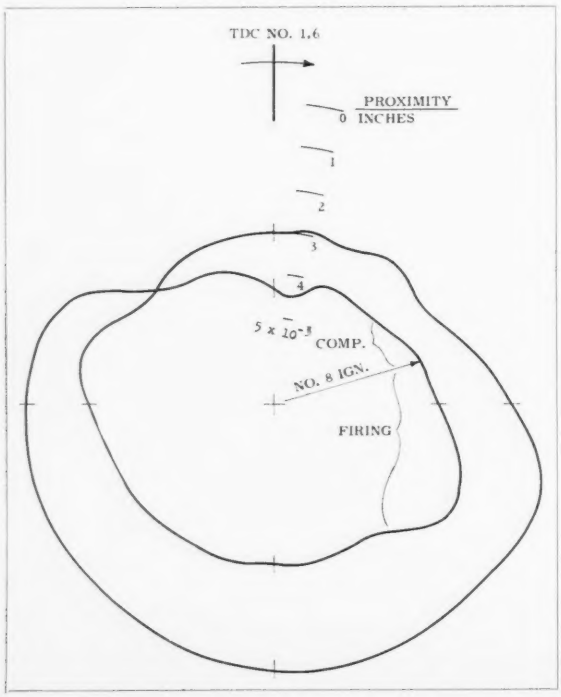

Fig. 7. Distance vs. Rotation.

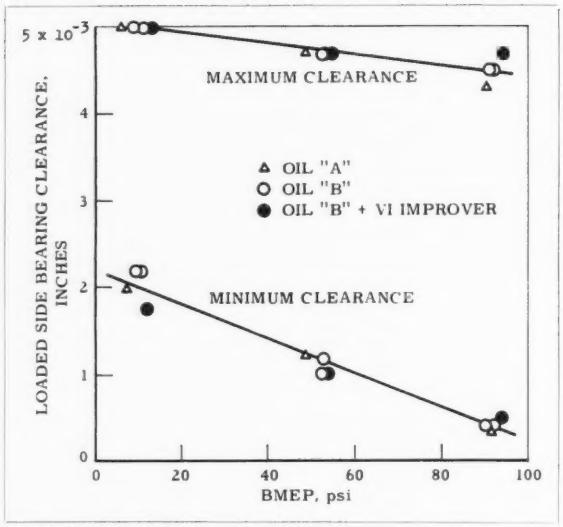

Fig. 8. Effects of Power Level and Oil.

thermal equilibria within the bearing. Thus, an increase in viscosity supplied increases frictional heating. The heating, in turn, results in increased oil flow and therefore greater heat rejection. Thermal balance is apparently achieved at a constant viscosity within the bearing for a given load condition.

These observations in a crankshaft main bearing of an actual operating engine indicate that the bearing clearance, while directly reflecting the gas load on the pistons, is insensitive to lubricant viscosity changes whether contributed by polymeric thickeners or by changes in the petroleum base stocks. Thus, neither the non-Newtonian character nor any permanent viscosity loss, from polymer degradation, of polymeric lubricating oils would be expected to lead to bearing damage as long as the oil supply is sufficient to maintain the temperature equilibrium.

AUTHOR'S NOTE:

It is of interest that the rather controversial results reported in this paper are in good agreement with subsequent observations by G. B. DuBois, F. W. Ocvirk, and R. L. Wehe. Their experimental observations that bearing oil film thickness with polymeric oils is greater, for a given condition, than would be predicted by calculation are presented in NASA Technical Note D427, May 1960, entitled "Study of Effect of a Non-Newtonian Oil on Friction and Eccentricity Ratio of a Plain Journal Bearing".

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The Development of a Test for Evaluating GRINDING FLUIDS

by R. D. Halverstadt,
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Evaluation of grinding oils is a difficult and uncertain process. Several methods were tried and a brief summary of each is reported herein. The paper briefly reports the results of several different approaches and concludes with recommendations for a simplified method.

INTRODUCTION

Grinding and cutting fluid evaluation has become increasingly important with the advent of the high temperature alloys. The correct fluid can assist in optimizing the processes which are at best difficult and expensive.

Grinding the new materials is usually accompanied by excessive wheel wear, poor surface finish and mechanical or metallurgical damage. Thus, it is important that a definite set of selective or evaluation criteria be developed.

This paper describes several different tests which were used to evaluate the fluids. The simplified test which has been evolved from the more complicated studies is described in detail.

There is no direct connection between any of the tests described herein. There is a relationship, however, because the facts learned from one evaluation were applied to the next study. It is to be stressed, therefore, that each set of studies described herein are in themselves complete stories and not necessarily related quantitatively to the other material contained in this paper.

This paper is not intended to be a detailed analysis of each of the types of testing described. Each study described herein is easily the subject of a complete paper. This paper is merely a cosmetic review of results which led to the final test which was evolved to be used in evaluation of oils and other fluids for use on high or medium temperature alloy materials. The text is divided into several distinct sections each with its own conclusions. Thus, parts A, B, and C are each cursory descriptions of a complete study. Part D is a description of the final test which has been evolved.

A. Oil vs. Synthetic Water Emulsions on SAE 4340. 321 S.S. and A-286

Test Procedure

In this study, three different materials were used with three different grinding fluids. Under identical conditions each fluid was tested with each material so that results could be judged on a direct comparative basis. Test bars 12 in. x 4½ in. of surface were made from each material. One cubic inch of material was ground from each bar on each test using a downfeed of .002 in. per pass with a crossfeed of .030 in. per pass. Thus, each test removed approximately 20 mils from the surface of the bar at which point the test was discontinued. The wheel was measured before and after the test and grinding ratio was calculated and the surface finish of the bar was measured. Each test variable was evaluated versus all other test variables in a statistically designed test.

Test Variables

Materials	SAE 4340	A-286 (commercial designation)	321 SS
Fluids	Oil-low sulphur	Synthetic water Emulsion #1	Synthetic water Emulsion #2
Table Speeds	10 SFPM	20 SFPM	30 SFPM
Wheel Hardness	G	I	K

Test Results

The test results were conclusive in that they clearly separated the relative results between the three different materials and two different basic types of fluid. The complete test results are presented in Table 1. Fig. 1 shows the result with the "I" hardness wheel at 20 SFPM table speed. A study of Table 1 will show that this is the general trend of results which were obtained in the complete evaluation. This points up clearly that there are great differences in wear ratios be-

Presented at the 15th ASLE Annual Meeting, Cincinnati, Ohio, April, 1960

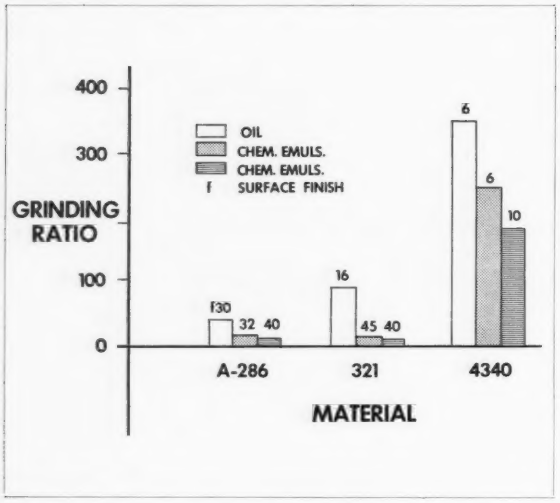

Fig. 1. "G" ratio vs. material—3 different alloys with 3 different fluids.

TABLE 1.—THREE FLUIDS ON THREE DIFFERENT MATERIALS (STUDY A)

Surface grinding on 4.5 in. x 12 in. surface. Wheel speed 6000 SFPM depth .002 in. crossfeed .030 in.

			A-286 STAINLESS STEEL		321 STAINLESS STEEL		4340 STEEL	
FLUID OIL	WHEEL GRADE	TABLE SPEED SFPM	FINISH A.A.	"G"	FINISH A.A.	"G"	FINISH A.A.	"G"
1	G	10	38	25	32	80	6	320
2	G	20	38	21	32	66	6	260
3	G	30	38	13	32	43	6	170
4	I	10	30	35	16	112	6	440
5	I	20	30	29	16	92	6	360
6	I	30	30	19	16	60	6	240
7	K	10	32	17	12	55	16	220
8	K	20	32	14	12	45	16	180
9	K	30	32	9	12	29	16	120
FLUID Synthetic #1								
1	G	10	38	4.6	70	3.6	4	230
2	G	20	38	3.3	70	3.0	4	190
3	G	30	38	2.1	70	1.9	4	120
4	I	10	32	5.6	45	5.0	6	320
5	I	20	32	4.6	45	4.1	6	260
6	I	30	32	3.0	45	2.7	6	170
7	K	10	60	2.8	60	2.5	16	160
8	K	20	60	2.3	60	2.0	16	130
9	K	30	60	1.5	60	1.3	16	85
FLUID Synthetic #2								
1	G	10	40	3.2	60	3.4	4	180
2	G	20	40	2.6	60	3.2	4	140
3	G	30	40	1.7	60	2.1	4	93
4	I	10	40	4.4	40	5.5	10	240
5	I	20	40	3.6	40	4.5	10	200
6	I	30	40	2.3	40	3.0	10	130
7	K	10	50	2.2	35	2.7	11	120
8	K	20	50	1.8	50	2.2	11	99
9	K	30	50	1.2	35	1.4	11	64

tween the stainless family and martensitic Alloys such as SAE 4340. The grinding ratio difference of over 10:1 under identical conditions shows that wheel wear in grinding the so-called "super alloys" is a serious problem.

The effect of the fluid is also demonstrated on the two stainless materials. The oil gave a much superior "G" ratio than did the two synthetic emulsions. All three fluids yielded about the same surface finish on A-286 but the oil is superior with respect to grinding ratio. On 321 SS the oil yielded the best surface finish and by far the highest "G" ratios. The SAE 4340 results were good with all three fluids from a standpoint of both surface finish and "G" ratio.

A discussion of the grinding ratio is in order at this point because many people attribute too much importance to this parameter. It is very important on some applications, but of little importance on many others.

In fact, the advantage in getting extremely high "G" ratios is nonexistent on most jobs. Fig. 2 clearly illustrates this point. This figure shows qualitatively the effect of the actual grinding rate as a function of "G" ratio for several different wheel sizes. On jobs where a large wheel is used, "G" ratios of 10 allow fairly high grinding rates. However, with a 1 in. dia wheel the "G" ratios must be much higher in order to approach a reasonable rate of stock removal. Thus, striving for high "G" ratios is pointless on some operations while extremely important on others.

Conclusions

(a) Oil yields "G" ratios superior to the synthetic emulsions by factors of the general order of magnitude of 5:1 on the two stainless steels with much less effect on the martensitic material.

(b) Oil yields better surface finish than either of the synthetic emulsions.

(c) General results of all conditions were good with the 4340 but relatively poor with the other two materials.

(d) The test clearly points to different performances with the three different fluids.

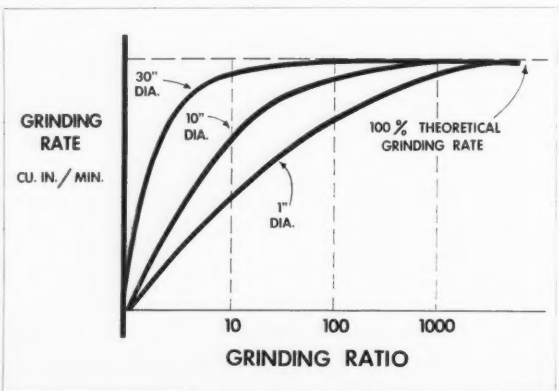

Fig. 2. Grinding rate vs. "G" ratio—the qualitative effect.

B. Oil vs. Two Ionizable Salts Evaluated on Titanium with Two Different Wheel Types

Test Procedure

This series of tests was conducted in a manner similar to the preceding test except that in addition to grinding ratio surface finish, measurements were taken on residual stress and photomicrographs of the cross section of the surfaces were made. Thus, a major emphasis was placed on surface damage. This was necessary at the time because of the tendency to surface "checking" during grinding and the creation of residual grinding stresses which lowered the endurance limit on titanium.

The stress-free samples used were approximately 0.100 in. thick with an area of approximately four square inches which were held flat during grinding by a mechanical fixture. The samples were ground at one set of conditions until .020 in. was removed from the surface. When the sample was released the deflection was measured between 2½ inch gauge points which gave a qualitative measure of the residual surface stress left in the sample. Thus, G-ratio surface finish, residual stress, and metallurgical damage were assessed on each sample. The test was designed and analyzed statistically.

The fluids used in this test were (A) an "active" commercial grinding oil; (B) Ba $(NO_3)_2$ mixed with water to a ½ molar solution; (C) Na NO_2 mixed with water to a ½ molar solution.

Two different methods of grinding titatium were evaluated namely (A) using silicon carbide wheels and (B) using Aluminum Oxide wheels. Each of the fluids was evaluated with each grinding method at three different wheel speeds and three different wheel hardness grades. The complete list of test variables is given below.

	SILICON CARBIDE	ALUMINUM OXIDE
Wheel	Commercial grade 80 grit 7 spacing, vitreous bonded in hardness grades I, K, M	Commercial grade 60 grit 6 spacing, vitreous bonded in grades I, K, M
Wheel Speeds	4000, 5000, 6000, SFPM	1500, 2000, 2500, SFPM
Table Speeds	30 SFPM	same
Down Feeds	0.0005″, 0.0015″, 0.0025″	same
Fluids	Active oil, Ba $(NO_3)_2$ (½ molar) Na (NO_2) (½ molar)	same

Test Results

Typical results with both methods are shown in Fig. 3 and 4. The aluminum oxide wheels yielded slightly better finishes and higher grinding ratios when the oil was used as the grinding fluid. The silicon carbide wheels yielded much higher grinding ratios and slightly higher finishes with the oil fluid. It is important to note that the two ionizable salts showed superior results when used with aluminum oxide wheels as compared to silicon carbide.

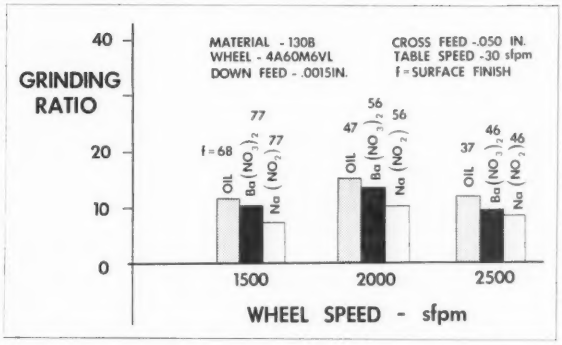

Fig. 3. "G" ratio vs. wheel speed and grinding fluid on 130-B titanium using aluminum oxide wheels.

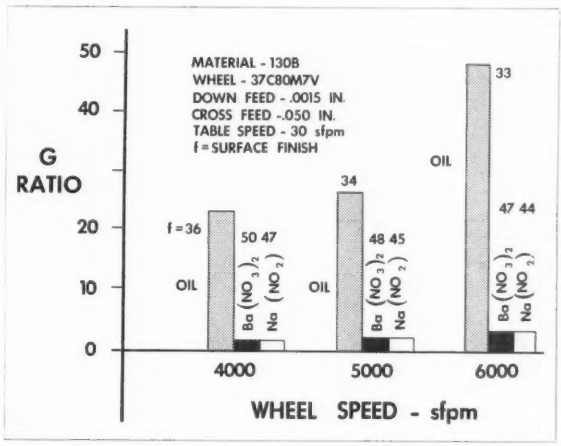

Fig. 4. "G" ratio vs. wheel speed and grinding fluid on 130-B titanium using silicon carbide wheels.

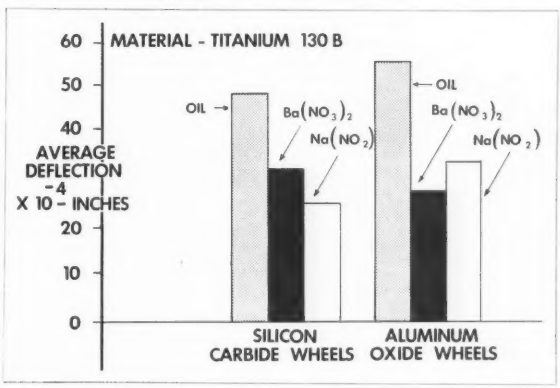

Fig. 5. Deflection vs. type of fluid with both types of grinding wheels.

Tables 2 and 3 and Fig. 5 show the results of the deflection tests. The tables show the average deflection resulting from the test as determined by statistical analysis of the test results. The deflections are given in inches as measured between two gauge points 2½

inches apart on the specimen. Positive readings indicate a tensile residual stress and it is significant to note that no compressive effects were found. The tables clearly show that the two ionizable salts are superior to the oil from a standpoint of residual stress. In fact, study of tables 2 and 3 shows that the salts are superior by a factor that varies from 1½ to 3.

Conclusions

1. The two salts were superior to the oil with aluminum oxide wheels when weighed against all criteria. Their benefit was especially significant in the effect on residual stress.
2. The oil was superior to the salts when used with silicon carbide wheels in terms of wear and surface finish. However, the salts were the best from a standpoint of residual stress and surface damage.
3. This type of test can clearly evaluate the benefits of one grinding fluid as compared to another.

TABLE 2.—Results of the Deflection Test Using Aluminum Oxide Wheels

	GRADE			
FLUID	I DEFLECTION	K DEFLECTION	M DEFLECTION	AVERAGE VALUE OF DEFLECTION
Oil	+58	+27	+82	+56 ±7.4
Ba $(NO_3)_2$	+23	+35	+27	+28 ±7.4
Na (NO_3)	+35	+30	+38	+34 ±7.4
Average Wheel Grade Value of Deflection	+38	+31	+49	+39 Experiment Avg. (±4.3 Error)

NOTE: The above figures are in inches deflection between 2½ gauge points multiplied by 10^4. Thus, a +34 indicates a deflection of .0034 in. bent to indicate an overall tensile stress effect.

TABLE 3.—Results of the Deflection Tests Using Silicon Carbide Wheels

	GRADE			
FLUID	I DEFLECTION	K DEFLECTION	M DEFLECTION	AVERAGE VALUE OF DEFLECTION
Oil	+42	+43	+60	+48 ±4.9 Error
Ba $(NO_3)_2$	+27	+33	+37	+32 ±3.9
Na (NO_3)	+22	+22	+32	+25 ±4.9
Average Wheel Grade Value	+30	+33	+43	+35 Experiment Avg. (±2.8 Error)

NOTE: The above figures are in inches deflection between 2½ gauge points multiplied by 10^4. Thus, a +34 indicates a deflection of .0034 in. bent to indicate an overall tensile stress effect.

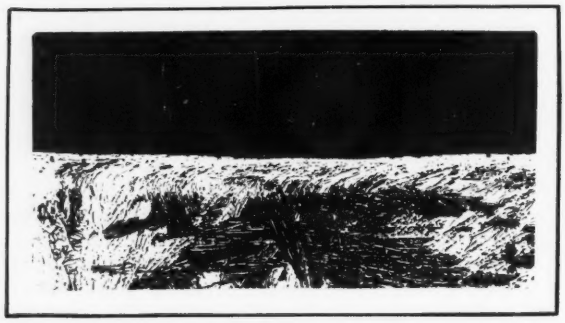

Fig. 6. Microstructure of 130-B titanium ground with aluminum oxide wheel and "oil. ' Wheel grade—4A60M6VL, Depth of cut—0.0025 in., Wheel Speed—2500 SFPM, Grinding Fluid—oil, 250 ×.

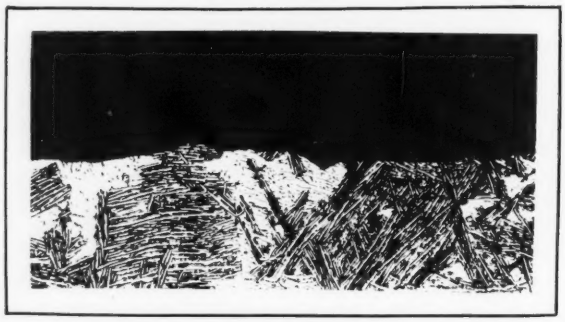

Fig. 7. Microstructure of 130-B titanium ground with aluminum oxide wheel and $Ba(NO_3)_2$. Wheel grade—4A60M6VL, Depth of cut—0.0025 in., Wheel Speed—2000 SFPM, Grinding Fluid $Ba(NO_3)_2$, 250 ×.

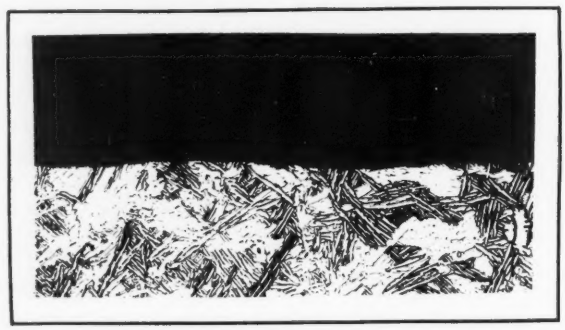

Fig. 8. Microstructure of 130-B titanium ground with silicon carbide wheel and oil. Wheel grade—37C80M7V, Depth of cut—0.0025 in., Wheel Speed—6000 SFPM, Grinding Fluid—oil, 250 ×.

C. A Series of Quantitative Residual Stress Studies

The tests developed thus far demonstrated that the grinding fluid had a a major effect on residual stress in the surface of the workpiece. At the same time the author's company had embarked on a program to improve endurance limit by control of surface residual stresses. Thus, over a period of several years many studies were made on the effect of fluid on surface residual stress with quantitative analysis by mechanical methods.

Test Procedure

The method used is fully described by Frisch and Thompson in (1) and reported on three alloys in (2) by Halverstadt. Essentially, the method consists of grinding thin stress-free strips of material held in a rigid fixture. When released, the samples bend as a result of the stress imparted by the grinding action. When these layers are removed from the stressed surfaces the specimens return toward their original position as each successive layer is removed. The change in deflection along with the known thickness of each layer can be used to calculate the average stress in the layer removed.

Some Typical Results

Two typical results on X-40 and U-500 materials are shown on Fig. 9 and 10 respectively. The superiority of grinding oil over the water base fluids is evident. This work corroborated the work of Letner (3) as mentioned by Halverstadt (2). Shown in Fig. 11, 12, 13 are results of Halverstadt which have been reported in (2). These data along with much other data have shown that the fluid is an important parameter in controlling surface residual stresses resulting from grinding.

Conclusions

1. Grinding fluid is an important parameter in controlling residual grinding stresses.
2. Any test designed to evaluate fluids should include the study of the effect on the surface of the material if it is to be used on a critically stressed part.

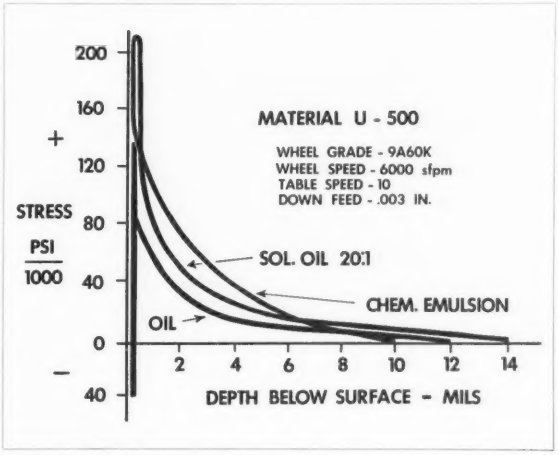

Fig. 10. Residual stress vs. depth below surface on U-500 material.

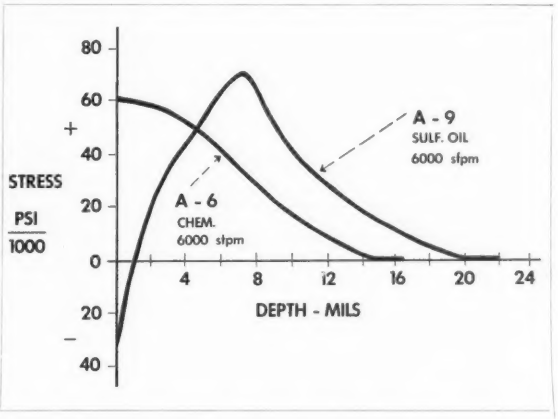

Fig. 11. Residual stress vs. depth below surface on A-286 material. (Ground at high wheel speed.)

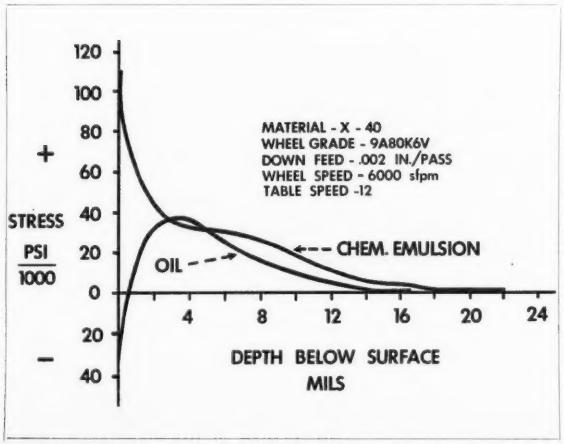

Fig. 9. Residual stress vs. depth below surface on X-40 material.

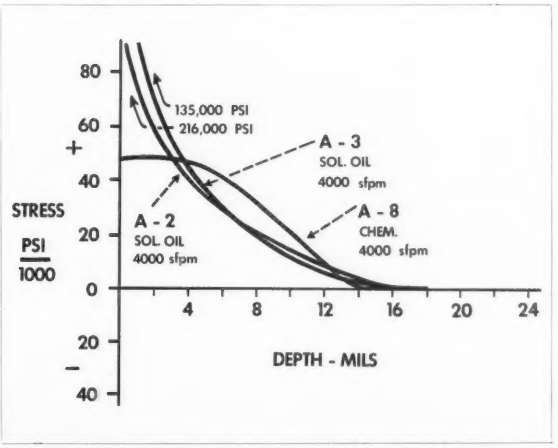

Fig. 12. Residual stress vs. depth below surface on A-286 material. (Ground at intermediate wheel speed.)

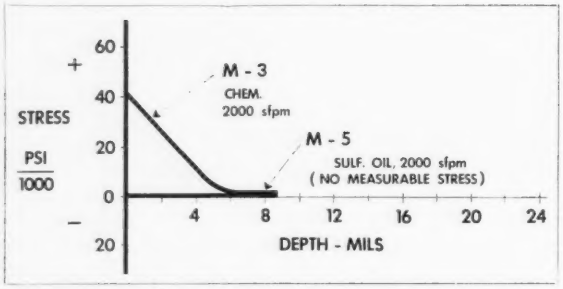

Fig. 13. Residual Stress vs. depth below surface on M-252 material. (Ground at low wheel speed.)

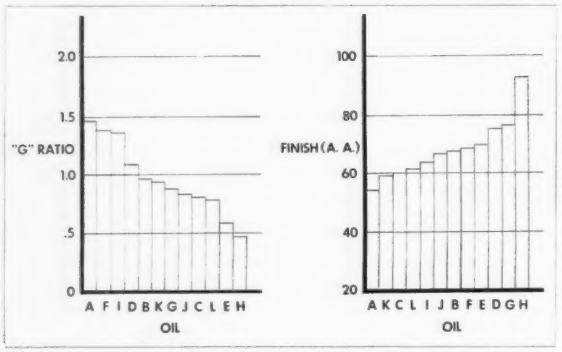

Fig. 14. "G" ratio and finish vs. type of oil.

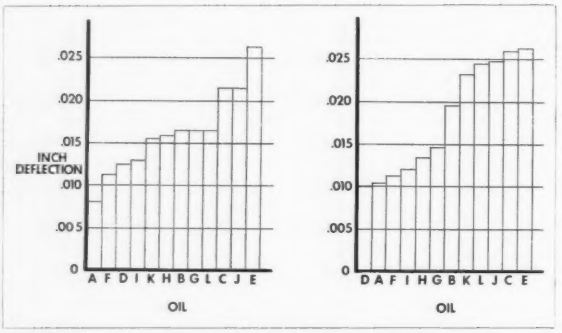

Fig. 15. Deflection of sample vs. type of oil. (Left) After rough grinding; (right) after finish grinding.

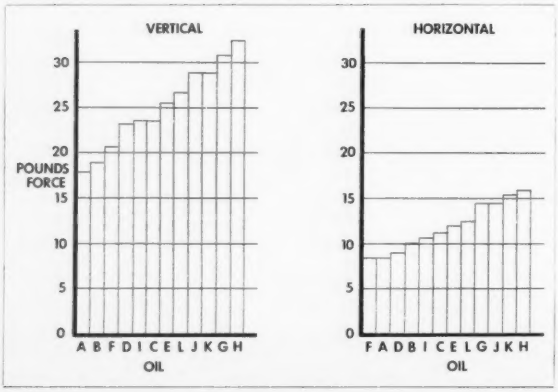

Fig. 16. Horizontal and vertical grinding force vs. type of oil.

D. Simplified Production Test

All of the foregoing results have shown that each of the measurements can be used as an important parameter in evaluating grinding fluids. However, such tests are extremely expensive and can be conducted successfully only in a research laboratory. A simplified test which could quickly evaluate the effect of a fluid on all of the important factors was needed. Thus, the following test procedure was evolved.

Test Procedure

The samples used in this test were samples of "stress-free" A-286 material ½ in. wide x ¼ in. thick by 6 in. long. The samples were held in a rigid fixture by mechanical means. The grinding wheel was plunged directly down into the sample. The specimen was ground from the original .200 in. thickness to a thickness of .100 in. At that time the test was concluded. Each set of conditions was used to make three samples from which the average value was used. The first series of studies used 12 different commercially available grinding fluids which were the only variables employed in the test. The constant conditions were wheel speed—4000 SFPM, work speed—15 SFPM, wheel grade—9A60K6VL, material A-286 solution treated and aged to 30 Rc. Plungetype surface grinding was used in all tests with roughing done at .002 in. downfeed/pass with the final .005 in. removed at a downfeed of .001 in. per pass. The measurements which were made were as follows:

Grinding Ratio—This was calculated by measuring the wheel both before and after grinding along with the known volume of work ground.

Surface Finish—Measured with a commercial profilometer after both rough and finish grinding.

Grinding Force—A two component grinding dynamometer was used to measure both the horizontal and vertical forces with each oil. (This was performed under finish grinding conditions on a separate specimen.)

Residual Stress—Residual stress was measured as a function of the deflection of the sample between 5 in. gauge points after both rough and finish grinding.

Ranking Procedure

As mentioned above, three specimens were ground with each oil and the average was used for the value of each measurement parameter. Point values for each measurement were established with 1 indicating the most desirable result and 12 the poorest. Thus, on the oil that had the best "G" ratio a value of 2, etc. All of the factors were evaluated in the same manner. The ratings for each oil in each category were added together to determine a composite rating for each oil. Thus, if an oil had been first in each of the six categories the final composite point rating would have been 6 which would be the best score possible. Under the other extreme the poorest composite rating possible would have been 72. Using the composite ratings the oils then were ranked 1-12 in order of their standing in the composite result.

TABLE 4.—Composite Ratings
1 = most favorable

Oil	G Ratio	Deflection		Forces		Surface Finish	Total Point Value	Final Rating
		Rough Grind	Finish Grind	Vertical	Horizontal			
A	1	1	2	1	2	1	8	1
B	4	7	7	2	4	7	31	5
C	7	8	10	6	6	3	33	6
D	3	3	1	4	3	10	24	3
E	9	9	11	7	7	9	52	11
F	2	2	3	3	1	8	19	2
G	6	7	6	11	9	11	50	10
H	10	6	5	12	12	12	57	12
I	2	4	4	5	5	5	25	4
J	7	8	9	9	10	6	49	9
K	5	5	8	10	11	2	41	7
L	8	7	9	8	8	4	44	8

NOTE: The above numbers are the relative ratings of the 12 different oils which were evaluated using the final test which was evolved. Thus, the number 1 indicates it was best of the 12; 2 second best, etc.

The result of the composite rating system is shown in Fig. 4. It is interesting to note that the actual composite ratings varied from 8 through 57 as contrasted to possible extreme limits of 6 through 72. Oil "A" is actually 11 points better than its closest competitor oil "F." The distribution is much more uniform for the other 11 oils. These tests and consequent relative numerical ratings can be duplicated at any time on some new oil from which selection decisions can be made.

In practice, the type of rating system can be used to select and approve grinding oils. This particular test was used to select four oils for shop application. It is a positive system for evaluation and thus far has proven valid in actual shop production activities.

The test as outlined above is much cheaper than the earlier tests and can be used to arrive at a composits number for final evaluation. No doubt, it still leaves much to be desired but it does consider all important factors and is in a way much more comprehensive than the earlier complicated tests since all the evaluation is made on a single specimen. This general method could well become the standard method for comparing grinding fluids.

It is evident that a different work material would have to be selected for each general class of material. However, as long as the general reaction can be fitted into any one class the results should be valid criteria for selection. Thus, only several classes would be necessary to evaluate most of the commonly used engineering materials. For example, on martensitic steel two different hardness levels might be sufficient to evaluate fluids to be used on all martensitic steels.

Many will note that different types of grinding will cause completely different results and this is true. However, past work has shown that a result obtained with this procedure will be essentially proportional on this same material with other types of grinding. Thus, the work indicates that an oil that shows good results on this type test will also be effective on other types of grinding on this same material. It is recognized that the effect will not necessarily be to the same degree.

Conclusions

1. The simplified composite test and evaluation system is effective in evaluating and comparing grinding fluids on any one general class of material.
2. The system is more economical and comprehensive than any of the earlier tests described herein.
3. The results of such a test can be used as a reliable criteria of evaluation for selecting grinding fluids.

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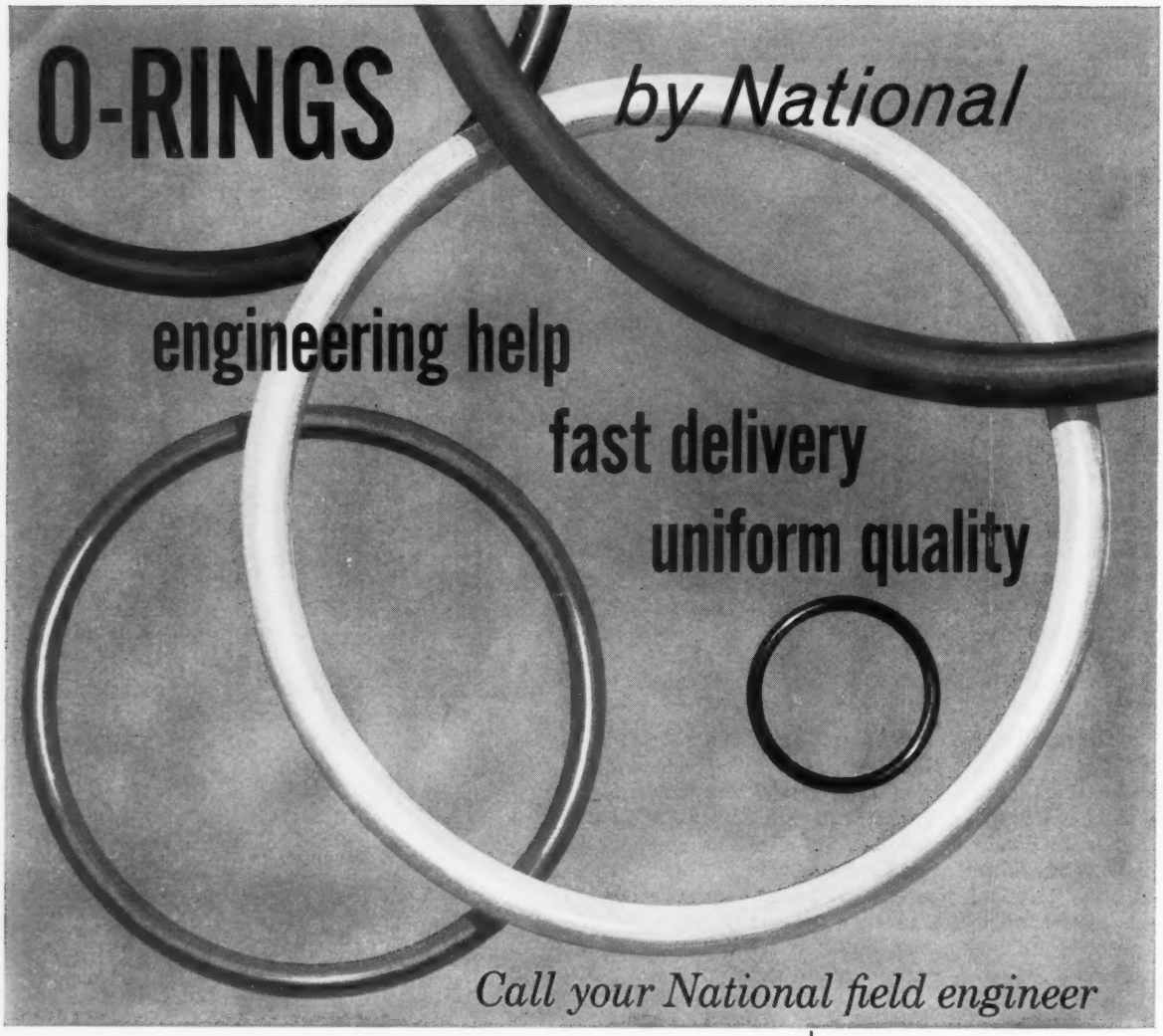
3. Halverstadt, R. D., "Analysis of Residual Stresses in Ground Surfaces of High Temperature Alloys," ASME Paper No. 57-SA-62.

ACKNOWLEDGMENT

The work prescribed herein is the results of the contributions of many people in the General Electric Company who collaborated with the author in planning and conducting the tests. The following people contributed in a major way to the various studies described herein:

D. A. Stewart, H. R. Soderstrom, F. B. Gabranski, A. Raeburn, D. R. Tague, K. W. Stalker and A. L. Luers.



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Lubrication Abstracts

Edited by W. E. Campbell

Selected literature compiled by members of the Technical Committees and Industry Councils of the Society.

H. S. Baldwin (Sheffield Div., Armco Steel Corp., Kansas City, Mo.), "The Use of Oil Film Bearings On A 12 In. Merchant Mill," *Iron & Steel Engr.*, v. 37, no. 6, 1960, 111-14, discussion pp. 114-116.

Eighty-four 12 in. bearings on a rolling mill system show favorable results from the standpoints of longevity and maintenance costs over a ten year period. The principal reason for achieving such a degree of success is that proper procedures are followed by the personnel responsible for bearing maintenance. Closed top housings also proved to be a decided advantage from the standpoints of rapidity of roll changes and safety. The procedures are: All horizontal roll housings of this fifteen stand mill are equipped with oil film bearings. Oil is fed to these bearings under pressure, from whence it drains to an oil reservoir on a lower level to be recirculated. Bearings are serviced in a bearing shop, where cleanliness keynotes the maintenance procedures. Painting the non-moving parts of the units make it quick and easy to clean this portion of the assembly. A system of rapid physical checks is developed to aid in the bench inspection of the units. Before a roll has left the bearing shop, bearings are checked for radial play between bushing and sleeve (journal), for contamination by foreign matter, and for evidence of entrained water in the lubricant. When excessive wear has developed, the worn bearings are moved back towards the roughing mill, where greater clearances are permissible. This plan resulted in prolonging the useful life of the bushing. Some of the slow speed roughing stand bearings ran for longer than eighteen months without dismantling. As a general rule bearings are reconditioned at approximately three month intervals. This procedure consists in cleaning, replacing seals, and replacing thrust bearings and bushings where necessary. A two-tank lubrication system is used for the mineral oil lubricant. The system is normally operated on one tank, with each tank on an alternate one week running basis. Contaminants are removed from the idle tank by heating the oil for a long enough period to precipitate most of the water and by centrifuging the balance of the entrained water. Disk type strainers are employed for removal of solid contaminants. Oil film bearings, in addition to the life and cost factors, are proved to have additional advantages when compared to other types of roll bearings. The ability of the bearing to hold the roll in an optimum location allows reasonably close dimensional tolerances to be maintained on rolled rounds and angles. A further advantage is the low starting torque. In installations of fabric lined water-lubricated bearings, for example, starting torque is found, by comparison, to be higher. In discussion, it is brought out that steel-backed babbit bearings have proved to be easier to maintain in rolling mills than aluminum bearings. *(Abstractor: William Rosenberg)*

L. N. Snell (Nuclear Power Plant Company, Great Britain), "Journal Bearings with Gas Lubrication," *Engineering*, v. 188, no. 4876, October 2, 1959, pp. 285-6.

This resume of "Pivoted-Pad Journal Bearings Lubricated by Gas," UKAEA (United Kingdom Atomic Energy Authority), Report No. IGR-R CA-285, obtainable from HMSO, 3S6D, describes experimental work on self-acting air-lubricated pivot-pad journal bearings. The author gives pressure profiles and load characteristics for 120 degree pivoted-pad journal bearings with different pivot positions. He compares results at low bearing numbers with published results for incompressible films. Two degrees of freedom of the pad in vibration and whirl of the rotor are described. System resonances due to the shaft mass and the stiffness of the film are described. *(Abstractor: W. A. Gross)*

R. Courtel and J. Larbre (Institut Francais du Petrole, Paris, France), "Analogy between Aqueous and Oil Medium in Case of Suspensions," *Revue de l'Institut Francais du Petrole et Annules des Combustibles Liquides*, v. 14, no. 12, 1959, 1763-1766 (in French).

It is shown that the problem of sustaining solid particles in detergent oils can be studied by the analogy between the well investigated electro-osmotic phenomenon of an aqueous medium and phenomena occurring in a hydrocarbon medium. *(Abstractor: George C. Lawrason)*

J. S. Ausman, "Torque Produced by Misalignment of Hydrodynamic Gas-Lubricated Journal Bearings," ASME—Paper n 59-Lub-3 for meeting Oct. 20-22, 1959, 6 p.

The angle of inclination as a perturbation parameter is used to obtain a first order analytic solution for misaligned air-lubricated journal bearings. When the eccentricity ratio is everywhere small enough, the results may be superimposed on those previously published by the author for the perfectly aligned journal bearing because the first order perturbation equations are linear in terms of the pressure. Pressure profiles for the misaligned bearing are odd functions whereas those for the aligned bearing are even. The misalignment torques given may be used to calculate film stiffness due to rotor inclination. *(Abstractor: W. A Gross)*

H. Kiene (Tractor Proving Ground—Darmstadt), "Tests with Lubricants of Viscosities SAE 80 and 90". ATZ v. 60, no. 8, 1958.

Prior to about 1955 truck transmissions generally used SAE 90 gear oils but since then the use of SAE 80 has crept in. To determine what advantages might accrue from using SAE 80 in farm tractor gearboxes, 3-hour tests are run on a 19 HP tractor having a 5-speed and reverse gearbox. Operation is in 2d speed at 5.12 km./hr. at rated engine speed, pulling a dynamometer on a cement road; drawbar pull and fuel consumption are measured. Outside temperature is 7-8C, motor oil temp start 29C, finish 40C; gear oil temp. SAE 80 start 13C, finish 29C; SAE 90 start 15.5C, finish 31C. The drawbar pulls are 910 kg and 895 kg respectively, or an increase in fuel efficiency of 1.65% for the thinner oil. This is attributed entirely to smaller churning losses. At an average load factor of 30% for farm tractors, the gain would be 5.5%. In other tests a 17 HP tractor pulls a trailer around a 94 km circuit on public roads at various load factors, all below 58%. Under the conditions of the tests the estimated annual fuel saving for a 17 HP tractor, used 1200 hours per year at 30% average load factor, is 22 U.S. gallons. *(Abstractor: E. A. Ryder)*

(Continued on page 137)

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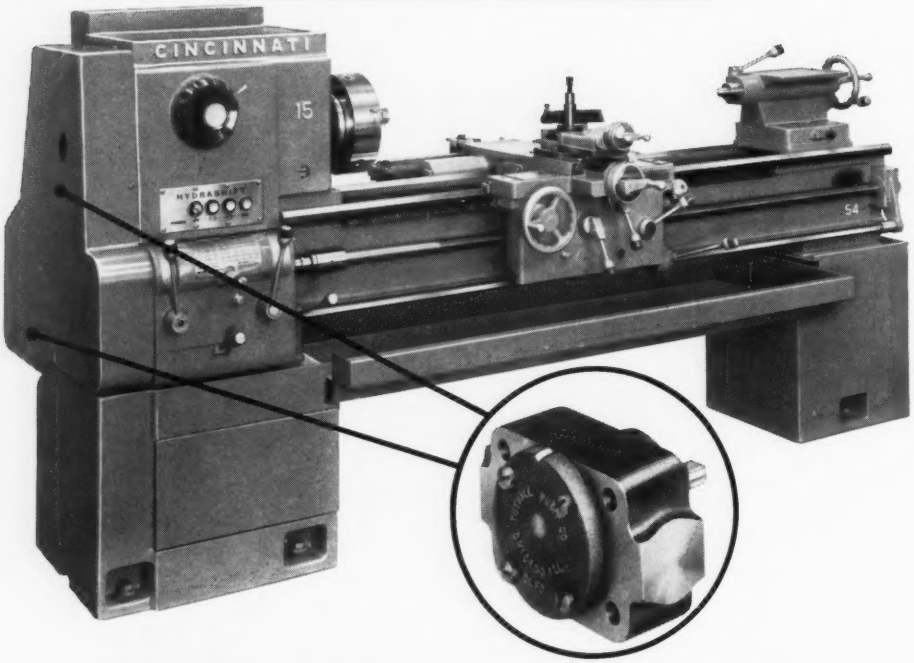

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LUBRICATION ABSTRACTS

(Continued from page 135)

K. K. Papok, K. N. Anisimov, B. S. Zuseva, and N. E. Kolobova, "Effect of Thio Derivatives of Unsaturated Phosphonic Acids on the Antioxidant Properties of Mineral Oil", Zhur Priklad. *Khim*, v. 32, no. 3, 1959, pp. 656-60.

The effects of thioesters of unsaturated phosphonic and thiophosphonic acids, and of diesters of alkyl thiovinylphosphonic acids on the antioxidant properties of mineral oil are studied. The thioesters of β-alkoxy (phenoxy) vinylphosphonic and β-alkoxy (phenoxy) vinyl thiophosphonic acids are effective antioxidants, the most powerful of these is the dithioethyl ester of β-ethoxyethoxy vinyl phosphonic acid. The antioxidant properties of esters of unsaturated phosphonic acids are greatly improved by the introduction of sulphur. The esters of alkyl thiovinylphosphonic acids do not increase thermo-oxidative stability nor diminish lacquer formation. *(Abstractor: Edward H. Loeser)*

J. Schutten (Laboratorium voor Massaspectrografie, Amsterdam, Netherlands), H. Baron, T. Van Der Hauw, and P. J. Van Deenen, "Dynamic Measurement of Film-Thickness in Gas Bearings," *Applied Sci. Research*, Sec. A, v. 7, no. 6, 1958, pp. 429-36.

A method is described for determining the mean position and the super-imposed vibration of one element of a gas-lubricated bearing relative to the other. The method involves the measurement of the capacitance change between the two members. The accuracy is approximately 0.1 μ m. Measurements may be made dynamically; therefore, the relative position of each member to the other is known at every moment. Use of the capacitance change for measuring film thickness and vibration in gas bearings is possible with gases which experience little or no change in dielectric constant over a wide range of temperature and pressure. One diagram of the circuit required for indicating displacement is presented. Another is presented which enables observation of the behavior of the shaft geometrical center by oscilloscope. *(Abstractor: P G. Smith)*

"Load Carrying Capacity Test for Oils—IAE Gear Machine" *J. Inst. Pet.* v. 45, no. 431, 1959, pp. 338-41.

This method, IP 166/60, Tentative, is a reissue of IP 166/58, Tentative, intended to define more exactly the degree of scuffing or scoring which constitutes failure; it contains descriptions of the various forms of tooth failure and a nomenclature of tooth parts. The test is for steel-on-steel spur gears and is run on the IAE 3¼" centers gear machine in steps of 5 min. run and 5 min. rest, with load increased by 305 lb per inch width of gear, for each run. Gears have hunting teeth. Testing one oil requires four pairs of gears. Three speeds and oil inlet temperatures are listed; choice of the conditions to be used is to be stated in oil specifications. Failure means scuffing or scoring on at least 60% of the tooth face area, and may occur at different loads on the approach and recess portions of the teeth; face area is between the pitchline and the tip of each tooth. The oil rating is the arithmetic mean of the failure loads determined on the separate trials. *(Abstractor: E. A. Ryder)*

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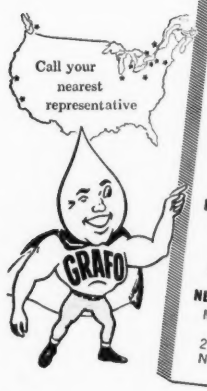

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Detroit Section's Past Presidents honored at January, 1961 meeting—Left to right—Back row: Mark Beardslee—1954-55; Wesley Wajtowicz—1960-61; Richard Leslie—1959-60; Claude Cowley—1958-59. Front row: John Swain—1952-53; Ralph Urso—1955-56; Edward Kelly—1950-51 and 1953-54; and Peter Uhl—1948-49.

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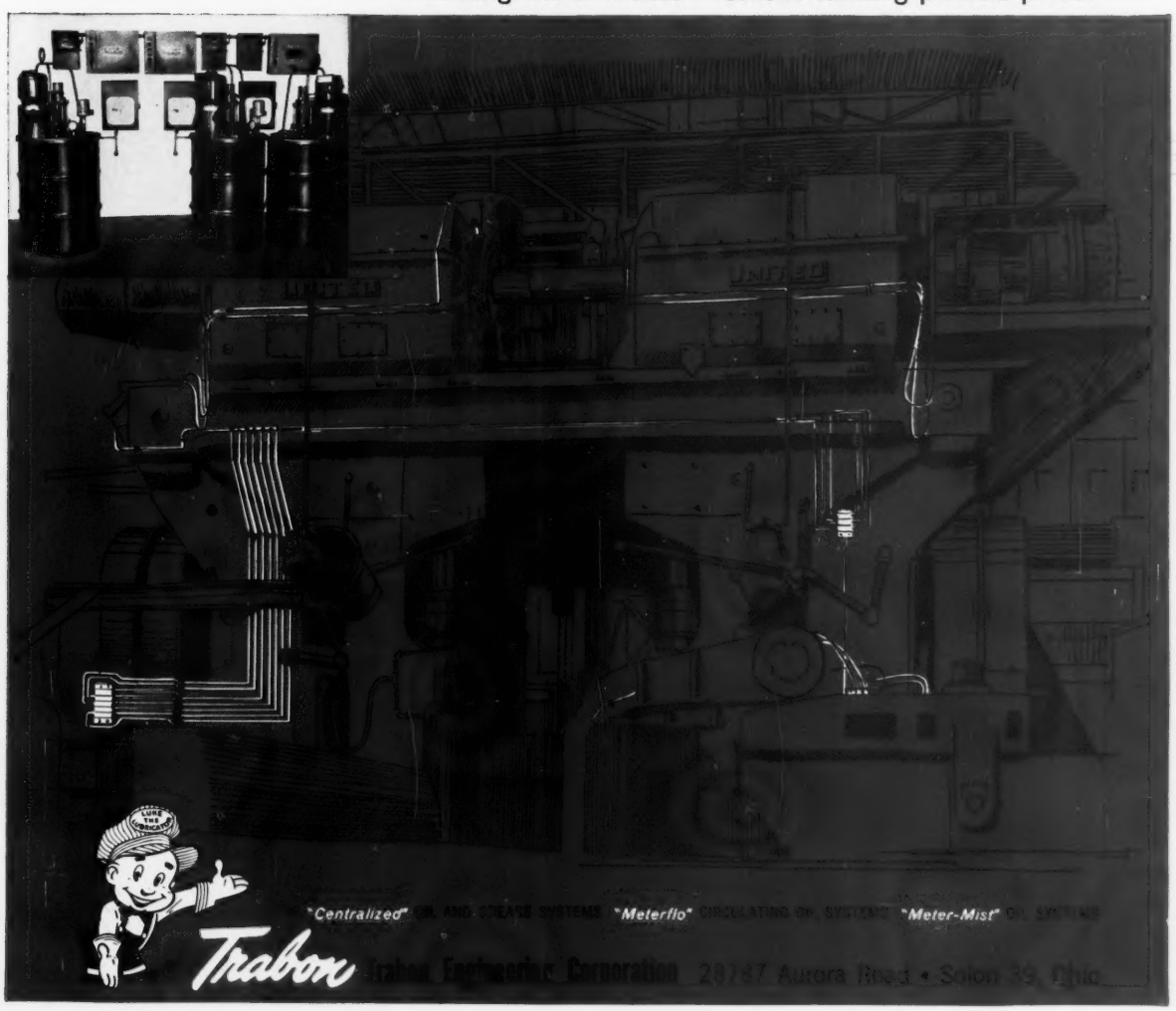

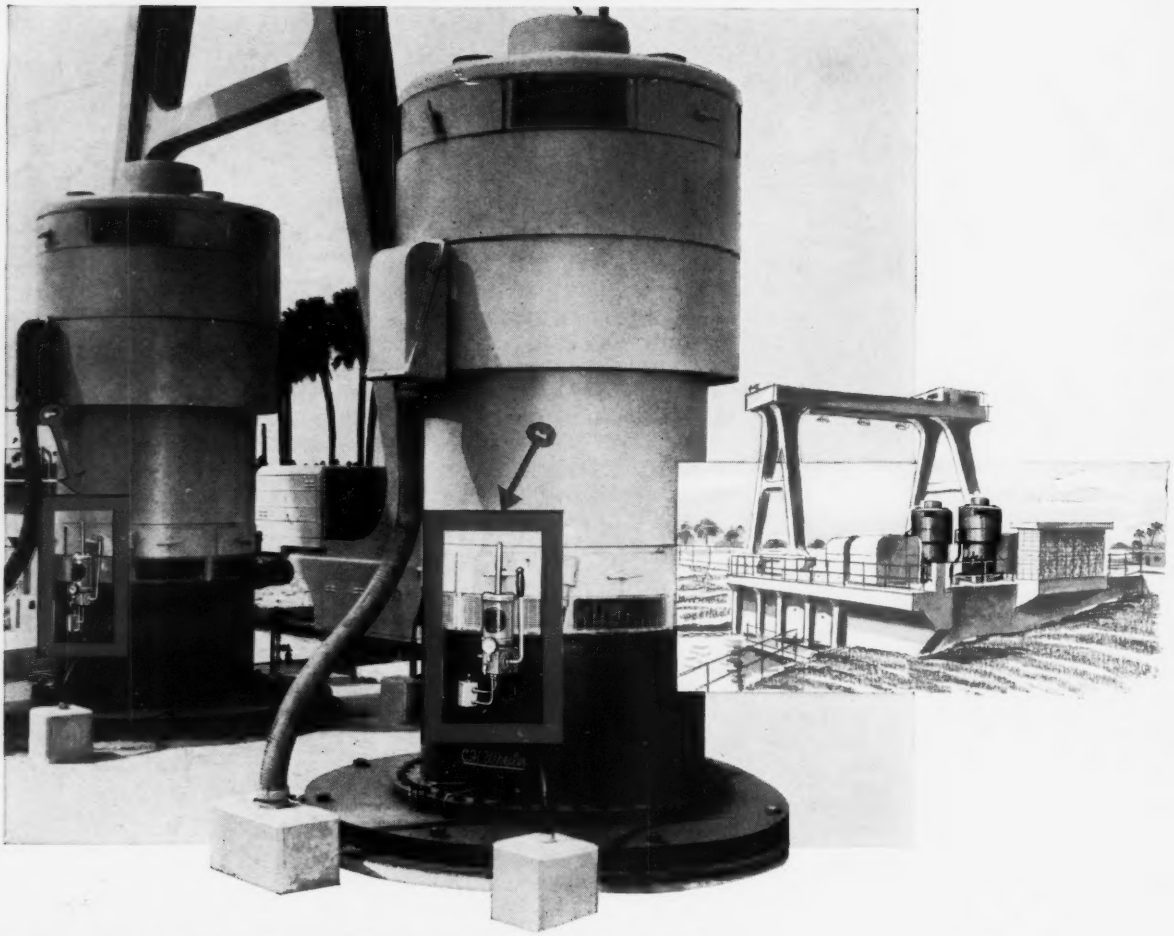

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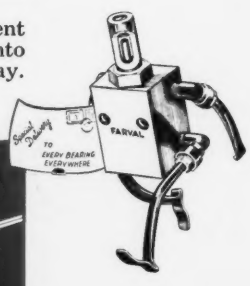

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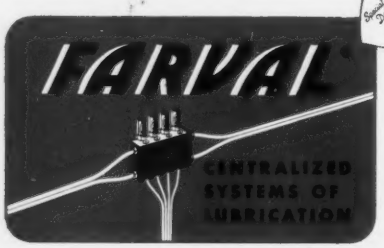

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